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XIX.

GRAPHIC METHOD FOR DETERMINING AND COUNTERBALANCING THE CENTRIFUGAL ACTION OF THE CONNECTING ROD.

By Passed Assistant Engineer A. B. Canaga, U. S. Navy.

The present demands for steam engines of high efficiency, constant and great rotative speed, has made it necessary to look carefully to the balancing of their moving parts. And in this paper it is proposed to show graphically and at a glance the extent of the most important causes disturbing the balance of an engine, and how these causes can be most nearly counteracted. Of course, it is impossible to perfectly balance a single engine, as will appear later, but a degree of approximation commercially satisfactory is attained by the process of trial and error. The following short cut is offered for consideration:

The parts of the engine, such as the crank and crank pin, having rotary motion alone, around other than the gravity axis, are counterbalanced by a weight placed diametrically opposite to and having a moment about the center of the crank shaft equal to the moment of the rotary parts about the same center,

acting from their common center of gravity. And these parts balanced for one speed are balanced for all. This part of the counter-weight can be determined readily, by an equation of moments so evident that the operation needs no explanation.

The next, and by far the most important, part of an engine requiring careful balancing, is the connecting rod. It has a motion of translation and rotation, and its center of gravity, if it coincide with the center of the crank pin, will travel in a circular path, and if it coincide with the center of the cross-head pin, its path will be a straight line; but if, as is generally the case, the center of gravity be between the crank and crosshead pins, it will travel in an elliptical path, the major axis of which will always be equal to the stroke of the piston, and the length of the minor axis will depend on the distance of the center of gravity from the center of the cross-head pin.

As the engine rotates, the constraining forces causing the center of gravity of the connecting rod to follow its elliptical path are opposed and balanced by a centrifugal force which acts normally to the path of the center of gravity. And this centrifugal force varies directly as the mass of the connecting rod, and the square of the velocity of the center of gravity, and inversely as the radius of curvature of the elliptical path of the center of gravity. The determination of the amount of this centrifugal force by analytical methods involves the solution of equations too cumbrous and full of transcendental functions to be of ready use to the engineer in balancing an engine. However, this centrifugal force can be easily represented graphically both in direction and amount, and from the diagram constructed to scale the weight of the counterbalance which shall counteract the part of the centrifugal action of the connecting rod which comes on the crank pin can be determined in the following manner:

Let W = weight of connecting rod in pounds.

" V = velocity of the center of gravity at any point of its path in feet per second.

Let r = radius of curvature in feet of the path at the corresponding point.

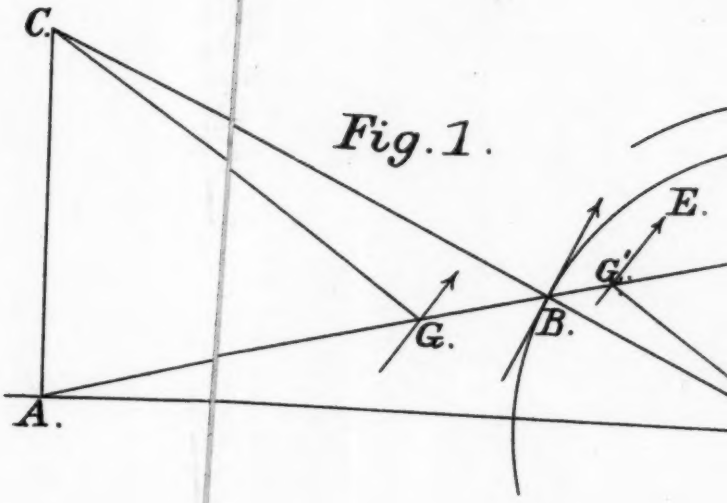
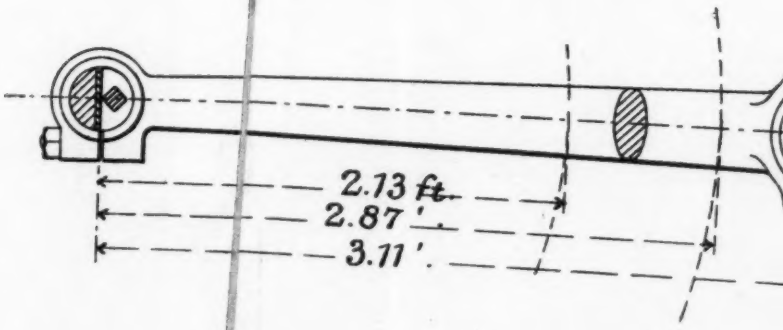
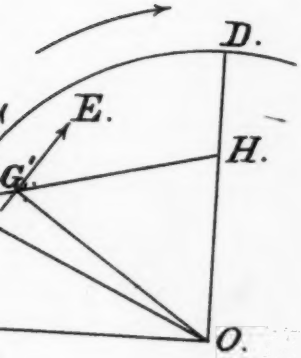


Fig. 1.

Fig. 3.

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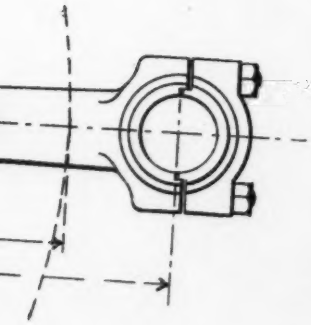
O.



Let $g = 32$.

Then the centrifugal force in pounds at the corresponding point will be,

$$C.F. = \frac{W \times V^2}{gr}. \qquad (1)$$

The graphical solution of equation 1 gives the amount of disturbing influence of the connecting rod in its rapid motion.

In order to find the velocity and direction of motion of the center of gravity of the connecting rod, the instantaneous axis of the latter must be known. In Fig. 1 the crank is represented by OB, and the connecting rod by AB. The direction of motion of A is along AO. Assume that the center of gravity of the connecting rod is at G, $\frac{3}{4}$ of its length from A. From A erect the perpendicular to meet the prolongation of OB in C, which will be the instantaneous axis of the connecting rod for the given position. At this instant the motion of A is perpendicular to AC, the motion of B is perpendicular to CB, the motion of G is perpendicular to CG, and in fact any point in the connecting rod AB is at the instant traveling at right angles to the line joining it to C. This fixes the direction of motion of the center of gravity of the connecting rod.

Draw OD perpendicular to OA, produce AB until it cuts OD in H, and lay off HG^1 from H equal to $\frac{3}{4}$ of HB. From kinematics,

$$AC : BC : GC :: \text{velocity of } A : \text{velocity of } B : \text{velocity of } G. \qquad (2)$$

In the triangles ABC and OHB, the side OH is parallel to AC, BH and BO are prolongations of AB and CB respectively, therefore the triangles are similar, and since OG^1 is parallel to CG,

$$AC : BC : GC :: OH : OB : OG^1. \qquad (3)$$

The first terms of proportions 2 and 3 being equal, gives,

$$OH : OB : OG^1 :: \text{velocity of } A : \text{velocity of } B : \text{velocity of } G. \qquad (4)$$

Now if OB be drawn to scale to represent the velocity in feet per second of the crank pin, OH and OG^1 will represent to scale the velocity in feet per second of the crosshead pin A, and the

center of gravity G of the connecting rod respectively. The direction of motion of the center of gravity of the connecting rod is represented by G^1E perpendicular to OG^1. Assuming that the crank pin has a uniform velocity in its path, the velocity of the center of gravity can be readily found.

The center of gravity is found by balancing the connecting rod across the edge of an iron bar. An $\frac{1}{8}$ to $\frac{1}{4}$-inch edge will locate the center of gravity sufficiently near for all purposes of balancing.

The r of equation 1 is the radius of curvature of the path of the center of gravity. From the form of this equation the centrifugal force will increase as V^2 grows larger, and decrease as r grows larger, and this variation of the centrifugal force is shown in Fig. 2.

In order to avoid generalities and show clearly the application of the following graphical scheme, the weights and dimensions for use are taken from a small engine of careful design:

Stroke of piston, ... 12 inches.
Diameter of piston, ... 6 inches.
Length of connecting rod, between centers, . $37\frac{5}{16}$ inches.
Distance from center of cross-head pin to center of gravity of connecting rod, . . . $25\frac{9}{16}$ inches.
Revolutions per minute, ... 300
Weight of connecting rod, . . . 45 pounds.
Velocity of crank pin, $\frac{1 \times \pi \times 300}{60} = 15.7$ feet per second.

Describe a circle from O, Fig. 2, as a center, and with a radius OD equal to the velocity of the crank pin to scale; in the present diagram $\frac{1}{8}$ inch equals 1 foot velocity, that is, the radius OD will be equal to $\frac{15.7}{8} = 1.96$ inches. Divide the circle into, say, 20 equal parts, D, 1, 2, &c. The more numerous the divisions the nearer will the solution approach the truth. However, 20 divisions will give a result quite near enough.

The ratio of the connecting rod to the crank is $\frac{37\frac{5}{16}}{6} = 6.22$. Lay off DA from D and $D''B$ from D'' equal 6.22 times OD, A

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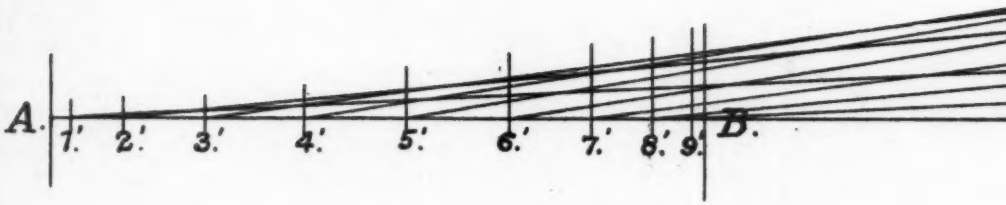
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Fig. 2.

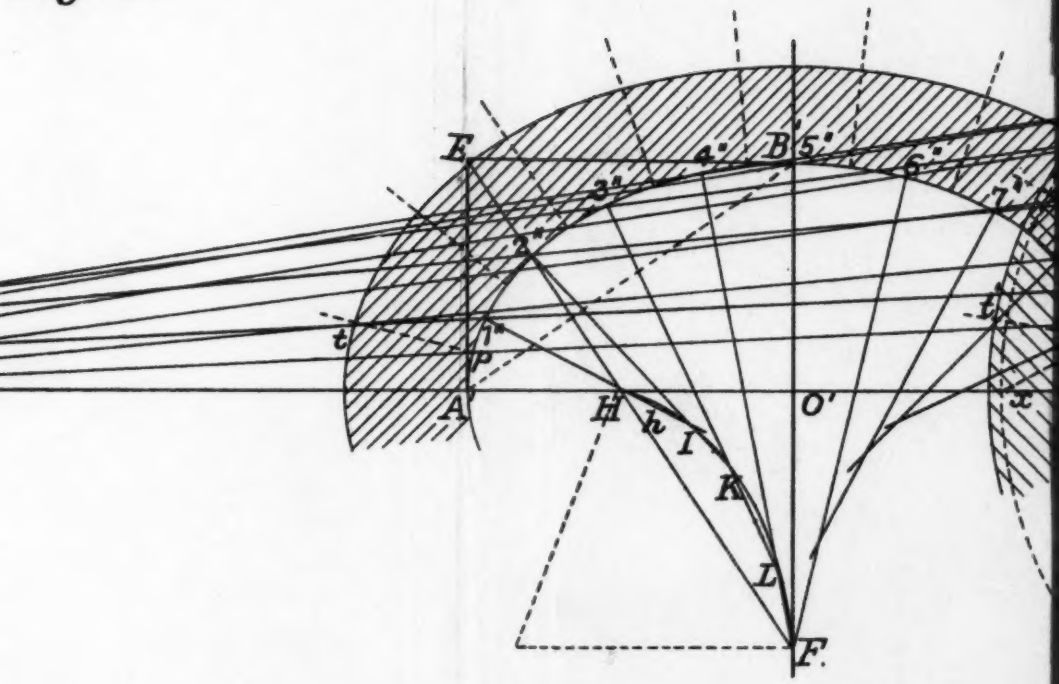
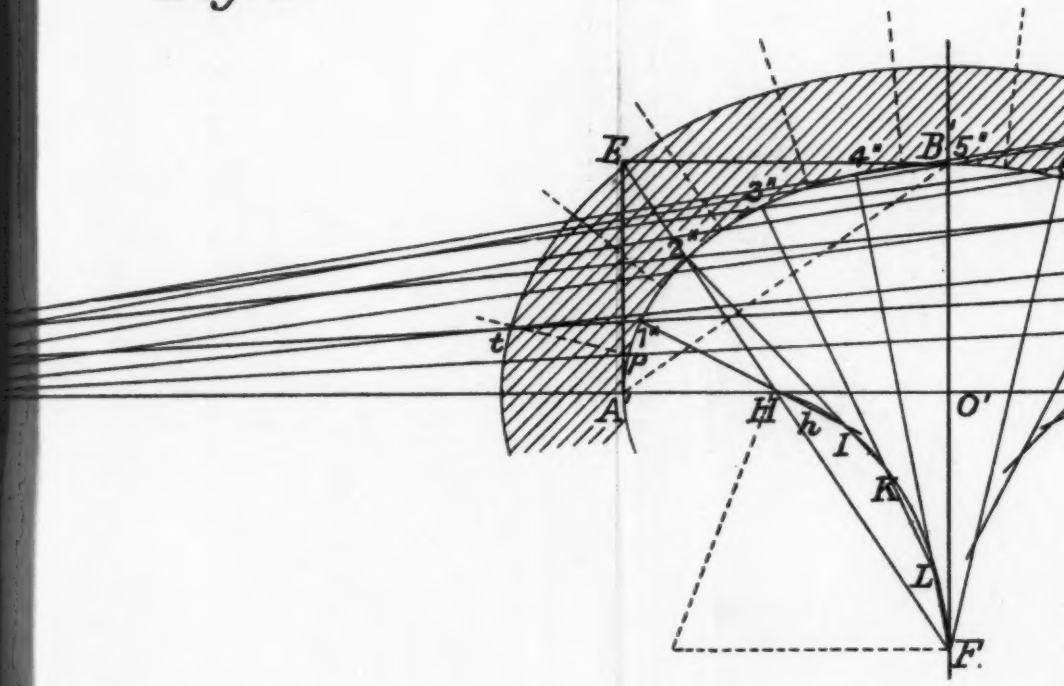
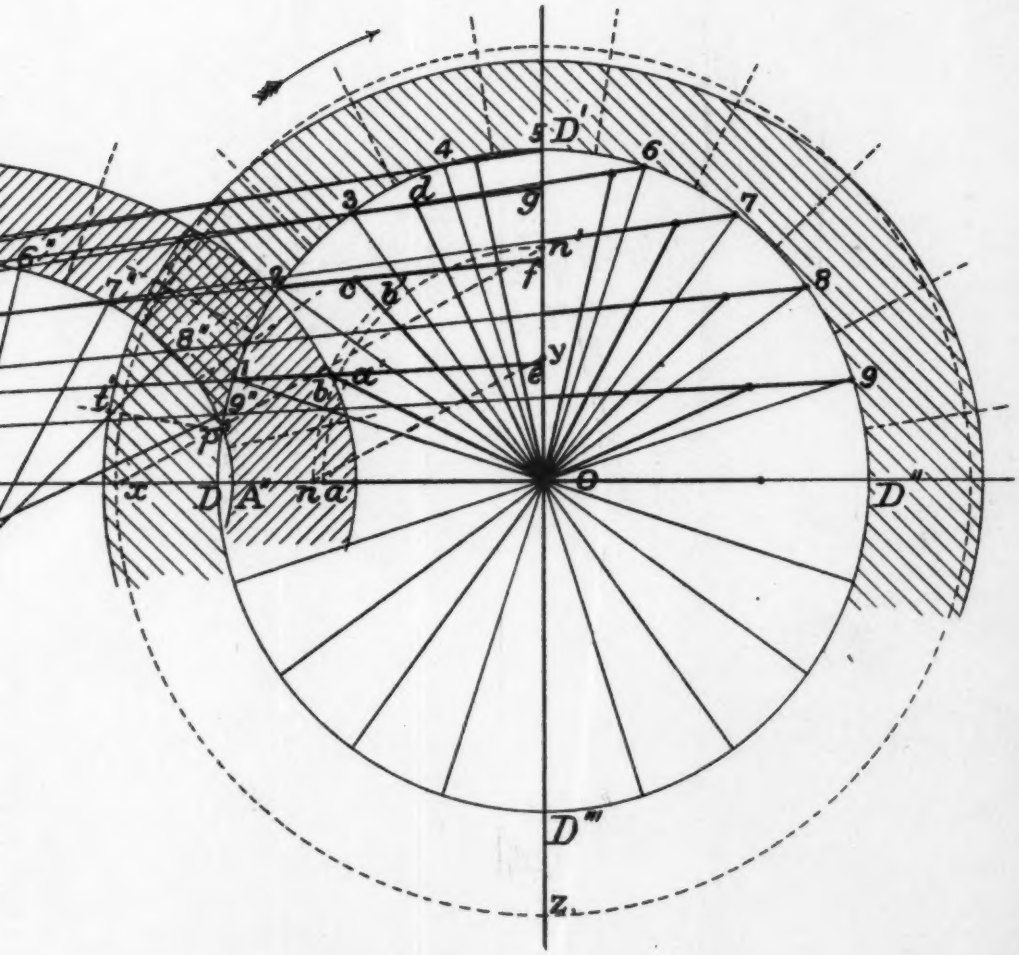


Fig. 2.

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and B will represent the two ends of the stroke of the piston, or more properly the stroke of the cross-head pin. And in the same manner find the positions of the cross-head pin $1'$, $2'$, $3'$, &c., corresponding to the positions of the crank pin 1, 2, 3, &c.

Connect $11'$, $22'$, $33'$, &c., producing the lines until they cut OD' or its prolongation in e, f, g, &c. Beginning at A, lay off AA', $1'1''$, $2'2''$, &c., equal to the distance that the center of gravity is from the cross-head pin. Joining the points A', $1''$, $2''$, $3''$, &c., gives the path of the center of gravity of the connecting rod for one revolution.

The distance of the center of gravity from the center of the cross-head pin is $25\frac{9}{16}$ inches, or $25\frac{9}{16} \div 37\frac{5}{16} = .685$ of the length of the connecting rod. Now from O, e, f, g, &c., lay off Oa, eb, fc, &c., equal to .685 of the distance OD, $e1$, $f2$, &c., respectively. Join a, b, c, &c., with O, then from what was shown in equation 4, Oa, Ob, Oc, &c., will represent to scale the velocity of the center of gravity of the connecting rod at A', $1''$, $2''$, &c. And, as was shown in connection with Fig. 1, the direction of motion will be at right angles to Oa, Ob, Oc, &c. While the crank pin traveled from D to 1, the velocity of the center of gravity was increased by the amount that Ob is greater than Oa. No material error will be made by taking the mean of Oa and Ob as the mean velocity of the center of gravity of the connecting rod while it passes from A' to $1''$.

The distance from O to midway between a' and b will give to scale this mean velocity to be used in the solution of equation 1.

To avoid very long ordinates in the diagram, 100 pounds will be considered the unit of weight and force. This assumption will not affect the truth of the equation, and will give $W = .45$ (of 100 pounds).

On the semi-major and semi-minor axes $O'A'$ and $O'B'$, Fig. 2, construct the rectangle $A'O'B'E$, and from E draw the line EF perpendicular to $A'B'$, the points H and F will give the centers of curvature for the ellipse at A' and B', respectively.

Through H and F pass an arc of a circle which will cut $O'A'$ at H and be tangent to $O'B'$ at F. This arc will approximate to the cusp, which is the locus of the centers of curvature for the

quadrant $A'B'$ of the ellipse. The limits of the radii of curvature for the other quadrants of the elliptical path may be found in the same manner, or the centers of curvature already found may be projected into the other quadrants. From $1''$ draw $1''H$ and produce the line to cut the arc of the circle at I, and rom $2''$ draw $2''I$ cutting the arc in K, and so on for each of the divisions of the ellipse. The lines $H1''$, $I2''$, $K3''$, &c., are the radii of curvature for the corresponding points.

Another and easier way to find these radii of curvature is to draw through $1''$ a line parallel to Ob, through $2''$ a line parallel, Oc, &c. The intersections of these lines will give the centers of curvature I, K, L, &c.

The path of the center of gravity of the connecting rod must be divided into the same number of parts as the path of the crank pin.

Beginning at A' with a weight equal to that of the connecting rod, and a velocity represented to scale by Oa, the tendency of this weight is to fly off along the tangent $A'E$, and the restraining effort for the instant acts through H; and this restraining effort is equal to the centrifugal force,

$$C.\ F. = \frac{W \times (Oa)^2}{g \times HA'}. \qquad (5)$$

But when the center of gravity arrives at $1''$ the velocity has increased to Ob and the radius of curvature has increased from HA' to $I1''$, and consequently the centrifugal force has changed to

$$(C.\ F.)^1 = \frac{W \times (Ob)^2}{g \times I1''} \qquad (6)$$

and so on for the complete revolution, the centrifugal force varies.

In order to show in one equation, and by one ordinate, the amount of centrifugal force from A' to $1''$, the mean of Oa and Ob is taken as the mean velocity, and the mean of HA' and $I1''$ as the mean radius of curvature, which gives,

$$C.\ F. = \frac{W \times (On)^2}{g \times h1''}. \qquad (7)$$

Inasmuch as equation 7 gives the mean centrifugal force of

the connecting rod while the center of gravity travels from A' to $1''$, the ordinate representing this centrifugal force should be laid off on a radial line passing through the middle point of the division $A'1''$ of the center of gravity orbit, which for convenience is made the base of ordinates.

Resuming equation 1, and making substitutions, gives,

$$C.\,F. = \frac{W \times V^2}{g \times r} = \frac{.45 \times V^2}{32 \times r}, \text{ or}$$

$$C.\,F. \times 71.11 \times r = V^2. \qquad (8)$$

In Fig. 2 the diameter DD'' represents one foot to crank orbit scale, while by the velocity scale it represents $2 \times 15.7 = 31.4$ feet; therefore r in equation 8 must be multiplied by $\frac{1}{31.4}$ in order that the value of the centrifugal force in 100 pounds may be given in one-eighths inches.

Equation 8 then becomes,

$$C.\,F. \times 71.11 \times \frac{r}{31.4} = V^2, \text{ or}$$

$$C.\,F. \times 2.25r = V^2. \qquad (9)$$

Equation 9 is now easily solved graphically by laying off On on OD and On' on OD', each equal to V and $Ox = 2.25r = 2.25\ h1''$. Now through n draw a line parallel to xn' and produce it to cut OD' in y, then the intercept Oy will represent to scale the centrifugal force in 100 pounds. The proof of this appears from the similar triangles Ony and Oxn', in which

$$Oy : On' :: On : Ox, \text{ or}$$

$$Oy \times Ox = (On)^2. \qquad (10)$$

Oy = centrifugal force, $On = On'$ = mean velocity of the center of gravity of the connecting rod, and $Ox = 2.25r = 2.25h1''$.

On the first radial ordinate lay off $pt = Oy$, which represents both the amount and direction of the action of the centrifugal force of the connecting rod, acting through its center of gravity.

The same operation gives the amount of the centrifugal force for each of the divisions of the center of gravity orbit.

If through the outer ends of the radial ordinates a curve be passed, it will show at a glance the *total* centrifugal action of the

connecting rod, each one-eighth inch representing 100 pounds centrifugal force.

The center of gravity of the connecting rod is kept to its elliptical path by the reaction of the crosshead and crank pins; thus it is seen that part of the centrifugal force represented by the radial ordinates is taken by the crosshead pin. This part must be deducted from the total centrifugal force; the remainder will then be taken by the crank pin; and this latter is the important part, to be counterbalanced by a weight opposite the crank pin.

If the connecting rod were at rest, then the statical pressures on the crosshead and crank pins would be inversely proportional to their distances from the center of gravity of the connecting rod. But the oscillatory character of the motion of the connecting rod causes its centrifugal action on the crosshead and crank pins to be inversely proportional to their distances from the center of percussion of the connecting rod.

The irregular shape of the connecting rod makes it impossible to calculate by mathematical formula the position of the center of percussion; however, it may be found by swinging the connecting rod as shown in Fig. 3. A semi-cylindrical block of wood faced with an iron plate forms the bearing for a knife edge, which is the angle of a bar of steel. The block of wood is to bring the knife edge to the center of the crosshead pin bearing. The connecting rod, as now supported, makes a pendulum which will, when started, swing for several minutes with very little variation in the length of its oscillation. If the oscillation be counted for five minutes a good average of their time in seconds will be obtained, then from Church's Mechanics, p. 119, and also from Weisbach's Mechanics, p. 662,

$$k = \frac{t}{\pi}\sqrt{gs}. \qquad (11)$$

k = radius of gyration of connecting rod.

t = time in seconds of a single swing.

$g = 32.2$, $\pi = 3.1416$.

s = distance in feet from center of crosshead pin, or center of suspension, to center of gravity of the connecting rod.

The values of t and s determined for the connecting rod under consideration are as follows:

$$t = 1.0875 \text{ seconds}, \; s = 2.133 \text{ feet},$$

then,

$$k = \frac{1.0875}{3.1416} \sqrt{32.2 \times 2.133} = 2.87 \text{ feet},$$

that is, if the connecting rod in practice simply oscillated as a pendulum, the center of percussion would be exactly 2.87 feet from the center of the crosshead pin. But the motion of the connecting rod being a combination of rotation and translation, the center of percussion, when the engine is on the dead centers, will be found 2.87 feet from the center of the crosshead pin, and when the crank and connecting rod are at right angles to each other, the center of percussion will coincide with the center of gravity of the connecting rod. In the present part of this investigation, the vertical component of centrifugal force is the more important, so that the center of percussion, 2.87 feet from the center of the crosshead pin, will be taken as giving a nearer approach to the truth. Then the amount of effort resulting from the centrifugal action of the connecting rod and coming on the crosshead and crank pins will be inversely as the distances of these pins from the center of percussion, that is,

C. F. on crank pin : *C. F.* on crosshead pin : : 2.87 : .24, or

$$\frac{C.\ F. \text{ on crank pin}}{C.\ F. \text{ on crosshead pin}} = \frac{2.87}{.24} = \frac{12}{1} \text{ or } \frac{1}{13}$$

of the centrifugal force of the connecting rod comes on the crosshead pin, and $\frac{12}{13}$ comes on the crank pin.

Now, using the crank-pin orbit as the base of ordinates, and laying off on the radial ordinates through the middle of $D1$ the distance $p't' = \frac{12}{13} pt$ and so on, laying off $\frac{12}{13}$ of each of the ordinates having the elliptical base on the corresponding radial ordinates on the crank pin orbit. A curve passed through the outer ends of these ordinates gives to scale a diagram of that part of

the centrifugal effort which comes on the crank pin during a revolution of the engine.

The diagram for the forward and backward strokes are exactly alike, so that the diagram for one stroke only is drawn.

The mean of the ordinates of this latter diagram gives the mean centrifugal force of the connecting rod to be counterbalanced by a weight placed directly opposite the crank pin. This mean centrifugal force is represented by the circle passing through z, $D'''z = 480$ pounds, to scale, $\frac{1}{8}$ inch $= 100$ pounds.

If the distance R of the counterweight from the center of the crank shaft be known, its weight in pounds can be found thus:

$$\text{Mean } C.\ F. = \frac{C.\ W. \times V^2}{g \times R}. \qquad (12)$$

$C.\ W.$ = counterweight in pounds.

R = distance of counterweight from center of crank shaft in feet.

V = velocity of counterweight in feet per second $= \frac{2\pi R \times 300}{60}$

$= 31.4\ R$.

Mean $C.\ F. = Dz = 480$ pounds, from Fig. 2.

Making these substitutions in equation 12 and transposing, gives,

$$C.\ W. = \frac{480 \times 32 \times R}{(31.4\ R)^2} = \frac{15.55}{R}. \qquad (13)$$

Equation 13 gives the weight of the part of the counterbalance for the connecting rod at the given speed.

Putting the counterweight the same distance as the crank pin is from the center of the crank shaft, that is, $\frac{1}{2}$ foot, gives equation 13,

$$C.\ W. = \frac{15.55}{\frac{1}{2}} = 31.1 \text{ pounds.}$$

In this particular engine, two-thirds of weight of connecting rod acting on the crank pin is to be balanced by counterweight. Northcott, in The Steam Engine, p. 116, says that one-half of the weight of the connecting rod is best treated as a rotating weight, and balanced by a counterweight placed on the shaft

opposite the crank pin. It is possible to make a connecting rod in which the center of gravity would be located so that only one-half of the weight of the connecting rod could be properly balanced by a weight opposite the crank pin, but the design would not be so good for high-speed engines as one more tapering from the crank-pin end to the crosshead end.

Equation 13 gives the *nearest* possible counterbalance for the centrifugal effort of the connecting rod. A glance at the diagram in Fig. 2 will show that the counterweight is too heavy for the connecting rod when the engine is on its half centers, so if it be desired to prevent entirely the centrifugal efforts from producing vertical vibrations, they must be made equal at the half centers.

It is also seen that the counterweight must be made heavier than equation 13 indicates, in order that the centrifugal efforts of the connecting rod and the counterbalance may be equal at the dead centers, thus preventing horizontal vibrations. The excess of the centrifugal effort at the dead centers should be taken up by compression, along with the inertia of the crosshead, piston and piston rod.

Care in compression is required, for the small clearance and short compression produce jarring effects in a way that is not at times suspected. To make clear the foregoing idea, the jarring effect of an 8-inch gun is much greater if its recoil be taken up in six inches instead of six feet.

The total weight to be counterbalanced in the small horizontal engine is made up of the crank, crank pin, eccentric, in the present case two-thirds of the weight of the connecting rod, and any other weight revolving on an axis not through its center of gravity.

As far as possible, the counterweights should be placed diametrically opposite the parts to be balanced, otherwise a disturbing couple will be established, increasing in effect as the speed increases.

If the center of gravity of the connecting rod be too near the crosshead pin, the centrifugal effort coming on this pin will produce unpleasant jarring by lifting the crosshead from the slides.

Excessive cushioning, beyond that necessary to take up the inertia of the reciprocating parts, will produce the same effect.

The influence of gravity on the connecting rod has been neglected in Fig. 2. Its effect would be to shorten the mean centrifugal ordinate for the forward stroke about 5 per cent., and lengthen the ordinate for the return stroke an equal amount. This would leave the mean ordinate for the complete revolution the same as before, and consequently the counterweight for nearest balancing would remain the same.

NOTE.—The path of the center of gravity in Fig. 2 approaches so closely to an ellipse that it is referred to as an elliptical instead of an oval path. In drawing the figure the instruments showed no difference between the curvature of the first and second quadrants. The principles governing centrifugal force hold regardless of the path, which may be any curve, and still,

$$\text{Centrifugal force} = \frac{W \times V^2}{g \times r}.$$

V, the velocity of the center of gravity, and r the radius of curvature of its path.

XX.

ERICSSON COMPOUND ENGINE AND BELLEVILLE BOILER.

By Chief Engineer Isherwood, U. S. Navy.

[Experiments made in the Delamater Iron Works, New York City, on a non-condensing, single-acting, tandem, compound steam engine designed by John Ericsson, and on the Belleville boiler supplying it with steam.]

The steam engine with which the following experiment was made was patented in one particular by John Ericsson, and a small example of it was constructed for him by the Delamater Iron Works, in the city of New York, and used there during 1888 and 1889 for experimental purposes by Mr. Ericsson up to the period of his death.

Many experiments were made with this engine to determine its economic efficiency with different piston speeds, different steam pressures, and different measures of expansion with which the steam was used; also, with and without condensation of the steam. These experiments were made with great accuracy, both the fuel and the feed water being carefully weighed, and indicator diagrams taken from each cylinder every half hour. Most unfortunately the data of all these experiments, except the one which is the subject of this paper, were consumed in an extensive conflagration at the Delamater Iron Works; and Mr. Ericsson's death soon after, combined with the breaking up of that establishment, prevented their repetition.

The experimental engine was supplied with steam from a Belleville boiler not made in the United States, but imported from the Belleville Works, in France. It was exactly one-half of the boiler in Mr. J. M. Forbes' steam yacht *Shearwater*, and an engraving of it is given opposite page 173 of the May issue,

1890, of THE JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS. As the fuel and feed water used were accurately weighed, and the other necessary data taken, the experiment with the engine enabled the economic vaporization of the boiler to be exactly ascertained under the experimental conditions.

There were two other experiments made with the boiler alone to ascertain its economic vaporization, and in these experiments (which were repetitional) the temperature of the feed water was greater, the pressure of the steam lower, and the water level in the boiler higher, than in the experiments with the engine in operation, all of which tended to increase the economic vaporization. The experiments, therefore, supposing the data to be exact, served to determine the economy resulting from the aggregate of these changes in the conditions.

ENGINE.

The engine is horizontal, single-acting, tandem and compound, the cylinders being placed with their axes in the same straight line, and without any distance between them, the bottom of the small cylinder and the top of the large cylinder being in contact. The large cylinder is between the small cylinder and the main shaft. Both cylinders are without steam jackets. The piston rod is in common for the pistons of both cylinders, which are secured to it. This rod passes steam tight, by means of metallic packing, through a long brass sleeve, partly in the bottom of the small cylinder and partly in the top of the large cylinder, but there is no piston rod stuffing box. Should there be any steam leak between the rod and the sleeve, it will be from the small cylinder into the large one, where it will be utilized on the large piston before being exhausted. The pistons are fitted with the Wheelock packing, placed in grooves turned in the pistons to receive it. Each piston has two of these grooves. The piston of the large cylinder has the piston rod on both sides, top and bottom. The part of the rod above the piston extends into the piston of the small cylinder, while the part below the piston extends to the crosshead of the engine, into which it is secured. The piston of the small cylinder has the piston rod on its bottom side only.

The steam from the boiler enters only the bottom (below the piston) of the small cylinder, the top being permanently open to the atmosphere when the steam is used without condensation, as in the case of the experiment in question, or into the condenser when the steam is used with condensation. The steam is admitted to the under side of the piston of the small cylinder by a flat rectangular plate valve, which acts also as the cut-off valve by means of the steam lap given to it. The seat of the valve is upon the small cylinder, and the valve is covered by a small chest into which the steam pipe delivers the boiler steam, the full pressure of which is upon the back of the valve. The steam passage is straight, and at its center, has only the length of the thickness of the metal of the cylinder. The top of the small cylinder is closed by a cover to prevent the access of dust; the communication between it and the atmosphere was made by a passage extending down the side of both cylinders to the opening for the discharge of the exhaust steam of the large cylinder. The exhaust valve of the small cylinder is a duplicate of the steam valve, being a rectangular plate, but has its seat on the back of the chest instead of on the cylinder. There are thus two valves to the bottom of the small cylinder. The small cylinder, its bottom, and the passage from the top, are in one casting; the two valve chests and the top cover are cast separately and bolted on.

The exhaust valve of the small cylinder is the steam valve of the large cylinder, the exhaust passage of the former cylinder being carried from the face of its exhaust valve to the top portion of the large cylinder, delivering therein the exhaust steam from the small cylinder upon the piston of the large cylinder.

Only the upper portion of the large cylinder (above the piston) is used, the lower portion (below the piston) being in permanent communication with the atmosphere when the steam is used without condensation, and with the condenser when the steam is used with condensation. The exhaust valve of the small cylinder being the steam valve of the large cylinder, and having no steam lap, the steam flows uninterruptedly from the small to the large cylinder during the entire stroke of the piston of the latter.

The pressure of the exhaust steam, is, of course, always on the back of the exhaust valve of the small cylinder. The bottom of the large cylinder is closed by a cover through which the piston rod passes by a stuffing box, and the communication with the exhaust passage is made through a side opening.

The exhaust valve of the large cylinder is at the top of that cylinder and is a rectangular plate having its seat upon the back of the valve chest instead of upon the cylinder. The exhaust passage extends from the face of the valve to the bottom of the large cylinder, where it communicates with the side passage to the atmosphere or to the condenser. The large cylinder, its top, and the exhaust passage at its side, are in one casting. The exhaust valve chest is in another, and the exhaust passage between the exhaust valve and the passage on the cylinder is in still another casting. The lower cover, is, of course, a separate casting.

The three valves of the engine are each worked by a separate eccentric—three eccentrics in all.

Both cylinders and all their steam connections were thoroughly covered with non-heat-conducting material.

The pistons and valves of both cylinders were carefully tested for steam leaks and found to be perfectly tight.

The steam pipe from the boiler to the cylinder was 47½ feet long, 2½ inches in outside diameter and 2¼ inches in inside diameter. It had six right-angled bends, and was well covered with non-heat-conducting material.

The exhaust pipe from the cylinder to the top was 41¾ feet long, 5 inches in outside diameter and 4¾ inches in inside diameter. It had four right-angled bends.

The following are the principal dimensions and proportions of the cylinders required to be known in connection with the experiment:

Number of cylinders	2
Diameter of the small cylinder	8¼ inches.
Diameter of the piston rod of the small cylinder	2 inches.
Net area of the piston of the small cylinder, exclusive of rod,	50.3146 square inches.
Stroke of the piston of the small cylinder	16 inches.

Space displacement of the piston of the small cylinder, per stroke	0.465876 cubic foot.
Space in clearance and steam passage at one end of small cylinder	0.045949 cubic foot.
Fraction which the space in the clearance and steam passage at one end of the small cylinder, is of the space displacement of the piston of that cylinder, per stroke	0.0986292
Diameter of the large cylinder	16 inches.
Diameter of the piston rod of the large cylinder	2 inches.
Net area of the piston of the large cylinder, exclusive of rod	197.9208 square inches.
Stroke of the piston of the large cylinder	16 inches.
Space displacement of the piston of the large cylinder per stroke	1.832600 cubic feet.
Space in clearance and steam passage at one end of large cylinder	0.140330 cubic feet.
Fraction which the space in the clearance and steam passage at one end of the large cylinder, is of the space displacement of the piston of that cylinder per stroke	0.0765743
Ratio which the net area of the piston of the large cylinder is to the net area of the piston of the small cylinder, the latter being taken as unity	3.933665
Ratio which the space displacement of the piston of the large cylinder per stroke, plus the space in the clearance and steam passage at one end of that cylinder, is to the space displacement of the piston of the small cylinder per stroke, plus the space in the clearance and steam passage at one end of that cylinder, the latter being taken as unity	3.854696
Clearance of the pistons of both cylinders	0.125 inch.

BOILER.

The boiler is Mr. Belleville's latest design. The ash-pit, the furnace, the tubes containing the heating surface, the uptake, the chimney, etc., are placed immediately above one another, and lie within the same horizontal rectangular area of the furnace and its walls.

The ash-pit is of sheet iron, with a mean depth of 18 inches below the upper surface of the grate bars. Its sides and back are box shaped, to support the brick masonry forming the sides and back of the furnace. The front of the ash-pit is closed by a sheet-iron damper removable by hand.

There is but one furnace. Its sides and back are composed of walls of brick masonry 11 inches thick, resting upon the

sheet-iron box-shaped sides and back of the ash-pit beneath. The front of the furnace is wholly occupied by the furnace doors and by the rectangular-sectioned pipe receiving the feed water from the steam drum for distribution to the tubes. The back brick wall of the furnace is 26 inches high. The side walls are higher, being 33½ inches high at the front and 39½ inches high at the back of the furnace. The grate bars are in one length, of 57.6 inches, and they decline 3 inches from the front to the back of the furnace. The breadth of the grate surface is 40 inches in clear of the side walls. There are two furnace doors, hinged on the sides and closing at the center. These doors open nearly the whole width of the furnace. The furnace doors are not perforated for the admission of air, but immediately above them is a small horizontal pipe, supplying steam to two $\frac{5}{64}$-inch diameter tuyeres for the admission of steam in jets to the furnace above the incandescent fuel. The office of these jets is to mechanically mix the gases of combustion as they rise from the coal on the grate, and thus perfect the combustion. This will undoubtedly lessen the smoke when bituminous coal is burned, but no appreciable economy results. The draught of the boiler is accelerated by the steam thus admitted to the furnace. These tuyeres were not used during any of the experiments.

Immediately above these tuyeres, and horizontally across the entire front of the boiler, extends a cast-steel rectangular-sectioned pipe, 5 inches square on the outside, of ¾-inch thick metal, and containing 4.16 square feet of heating surface measured on the exterior of three sides, and 3.33 square feet of heating surface measured on the interior of three sides. This pipe receives the feed water from the steam drum, and distributes it to the tubes composing the heating surface. In the vertical flat side of the pipe, fronting the fire room, are four wrought-iron plugs, of 2¾ inches diameter, held by a T-bolt and nut, one plug opposite the center of each element or group of tubes. By means of the openings closed by these plugs the interior of the pipe can be examined and cleaned. The pipe is supported at its ends on the brick side walls of the furnace, and upon its upper flat side rest the lower front series of boxes or chambers of cast steel ½-

inch thick, forming the bends or passages of communication between the lower tubes of the same element or group. The corresponding lower back series of boxes or chambers of cast steel ½-inch thick, forming the bends or passages of communication between the lower tubes at the back of the furnace, rest upon the furnace's back wall of brick masonry.

The communication between the top of the horizontal rectangular-sectioned pipe containing the feed water, and each lower front box of each element or group of tubes, is made by a nipple of 1½ inches least diameter in the clear. Through this opening of 1.767 square inches area passes all the feed water for one element or group of tubes, the interior cross area of the tubes being 9.965 square inches. This "throttling" of the feed water is for for the purpose of securing its equal distribution to each element or group of tubes.

Each lower box (four in number for the boiler) is divided into two parts by a horizontal partition; the lower part receives one end of the lowest tube, and the upper part receives one end of the two tubes immediaiely above, so that each lower box receives one end of three tubes. This vertical box has two vertical faces, and is 9½ inches wide, 11 inches high, and 6 inches broad. In its front side, facing the fire room, are four wrought-iron plugs of 2¾ inches diameter, three of them being placed with their centers in the axes of the three tubes. By means of the openings closed by these plugs, the interior of the boxes and of the tubes can be examined. Each of these boxes has a total outside area of 2.6 square feet, and a total inside area of 2.0 square feet.

All the other boxes, front and back, receive one end of two tubes, and the entire interior of each of these boxes is in common, forming the bend or communication between the tubes. The front boxes are of cast steel ½-inch thick, twenty-four in number for the boiler, arranged in six vertical rows, one over and resting upon the other. These boxes, like the lower four, have two vertical faces. They are 9½ inches wide, 5 inches high, and 5½ inches broad. Their front side, facing the fire room, has two wrought-iron plugs of 2¾ inches diameter, each

placed with its center in the axis of a tube. The centers of the two tubes ending in each of these boxes, are in the same horizontal line. By means of the openings closed by the plugs, the interior of the boxes and the tubes can be examined. Each of these boxes has a total outside area of 1.35 square feet, and a total inside area of 0.92 square foot.

The back boxes are of cast steel $\frac{1}{2}$ inch thick, thirty-two in number, and are shaped like ordinary bends, being cylinders of 5 inches external diameter bent to semicircles; the end of the two tubes are screwed into each flat end of the cylinder. The axis of each cylinder is 14 inches long. Each of these boxes has a total outside area of 1.527 square feet, and a total inside area of 1.222 square feet.

The tubes are sixty-four in number for the boiler. They are of wrought iron $\frac{3}{16}$-inch thick. Each tube is 3.937 inches or one-tenth of a metre in outside diameter; its inside diameter is 3.562 inches; and its length in the clear between the boxes is 62 inches. The exposed external surface of each tube is 5.32532 square feet, and the exposed internal surface is 4.81808 square feet.

The tubes of the upper row, four in number, come from the upper back boxes, and their front ends are screwed into four single upward cast-steel bends, each of which is fitted with a horizontal flanged cylindrical vertical nozzle. The front and back faces of these bends are vertical, and the front face of each has an opening of $2\frac{3}{4}$ inches diameter closed by a wrought-iron plug the center of which is in the line of the axis of the tube: on removing the plug the interior of the box and tube can be examined. A vertical pipe of 4 inches diameter is bolted by its flange to the horizontal flanged cylindrical nozzle of each box, and connects with the steam drum or steam collector, discharging the steam therein. Each of these boxes has a total outside area of 1.236 square feet, and a total inside area of 0.778 square foot.

The tubes and their connecting boxes are surrounded as a whole, with the exception of the front of the boiler, by a double casing of sheet iron; the interval between the two casings is 3

inches wide for the principal part, and 4½ inches wide for the remaining part, and is filled with mineral wool or some other non-heat-conducting substance. The inner casing rests upon the top of the walls of brick masonry, and the outer casing extends down the sides and in contact with these walls. The front of the tubes and their connecting boxes are closed by two doors, hinged on the sides and meeting at the center. The enclosure of the tubes is completed by a double arched covering extending over all, and filled in with mineral wool.

The tubes are arranged in four entirely distinct groups or elements, each of which is composed of sixteen tubes, the sixteen forming one continuous tube by means of the connecting chambers, boxes or bends at their ends. This one long tube, therefore, is a forward-and-backward coil, the constituent tube portions of which lie one immediately above the other, and side by side. All the water that enters the lowest tube of each element from the rectangularly sectioned pipe or feed-water collector across the front of the furnace, passes necessarily through the sixteen tubes composing the element, and emerges as steam and water from the highest tube into the steam drum or steam collector. The distance, horizontally, between the centers of the tubes of the same group is 4½ inches, and vertically 5½ inches. All the tubes incline a little upward in the direction of the current through them. The lowest tube proceeds from one side of the lower portion of the lowest front box to the corresponding back box or bend; this box forms a horizontal communication with the next tube which returns to and enters the upper portion of the lowest front box, diagonally opposite to where the lowest tube left that box. This upper portion forms a horizontal communication to the third tube which leaves the upper portion of the lower front box and proceeds to the second corresponding back box or bend forming the horizontal communication there with the fourth tube which returns to the front box next above the lowest front box; this second front box forming, like all the front boxes above it, and like all the back boxes, a horizontal communication with the succeeding tube. And so on, until the upper tube is reached, which, returning from the back to the front

of the furnace, delivers its contents into the steam drum or steam collector. All the steam generated in all the sixteen tubes of the same element, is delivered through the upper tube of that element, together with any water that may be entrained by that steam. The aggregate length of the tubes and bends of one element is 91½ feet. Mean height of each element, 45 inches over the tubes. Breadth of each element over boxes, 9½ inches.

As each element or set of tubes is entirely independent of the other elements, any one element can be removed from the boiler, and the two openings thus exposed being capped—one in the lower front box and the other just beneath the steam drum—the boiler can be used as before. And so on; any number of the elements except one could be removed without interfering with the action of the remainder. The efficiency of the boiler, however, would be correspondingly reduced; and the horizontal projection of the removed element or elements in the uptake should be closed by a plate of iron to prevent the escape of the gases of combustion without having come into contact with heating surface. Baffle plates are placed among the tubes to force these gases into contact with the whole of the heating surface, and to prevent them from taking a straight course to the chimney. The calorimeter, or draft opening between the tubes, is thus reduced to five-eighths of what is due to the aggregate horizontal spaces between the tubes.

The front boxes are separated from each other vertically by a space of ½ inch, the support being formed by chipping pieces projecting ¼ inch from the top of one box and from the bottom of the box over it. The back boxes are similarly separated vertically, and supported on each other by similar chipping pieces. The ends of the tubes are simply screwed into the boxes, and have a narrow screw packing ring to be set up independently, so as to tighten the screw joint on the outside by packing. The three lower tubes, having each an end in the lower front box, are cut in two at 4 inches from that box, and the two parts are united by a screw sleeve 4 inches long extending over them. By means of this device each element can be drawn out separately with rapidity, and without difficulty and independently of

the others, and without disturbance of any other part of the boiler, through the front doors for examination, cleaning, repair, or renewal of any part. This can be done with small hand tools, and upon the floor of the fire room, and by the ordinary engineering staff of a steamer, so that each element of the boiler being taken out in succession, all can be examined, cleaned, repaired and replaced, without involving any further detention of the vessel or loss of time than is required for taking out and replacing one element—a few hours at the most—or the boiler can be used without the element in question. The boxes being free to move on each other according as the tubes are expanded or contracted in length by variations of temperature, furnishes all the provision necessary for security in that respect. The position, and the number of the tubes in height, are such that they can be satisfactorily swept of soot in place without difficulty when the boiler is cold.

The steam drum or steam collector is a cylinder extending horizontally nearly across the front of the boiler, at the top. It is 18½ inches in outside diameter. The feed water is intended by Mr. Belleville to pass through this drum from end to end, and to be exposed during its passage to the temperature of the steam, which is maintained there sufficiently high to precipitate all the salts of lime (sulphates and carbonates) that the water may contain. The feed water being pumped into the steam drum at one end, is intended to flow along the bottom of the drum to the other end, whence it is to descend through a vertical pipe to the horizontal rectangularly cross-sectioned pipe from which the tubes of the different elements are supposed to receive their feed water. In descending through the vertical pipe, it carries along the precipated calcareous matter, depositing the latter by gravity in the bottom of a vessel, from which it is blown out by the pressure within. The top of this vessel communicates with the horizontal rectangularly cross-sectioned pipe, which thus receives feed water freed from salts of lime. The separation of these salts is thoroughly and almost instantly accomplished by a temperature of about 340 degrees Fahrenheit, corresponding to a saturated steam pressure of about 105 pounds per square inch above the atmos-

phere, and the precipitate is said to be expelled in the form of an incoherent granular slush. With this apparatus operating as intended, sea water could be used to feed the boiler without the slightest danger of scale forming upon the heating surface, but the iron would be instantly attacked by the sea water and rapidly destroyed by the consequent corrosion. The device is not original with Mr. Belleville, and can be applied to any boiler in which the steam pressure is not less than about 105 pounds per square inch above the atmosphere. It is, of course, no portion of the boiler proper, and could not be used with a sea-water water feed, either in whole or in part, on account of the destruction of the iron; and even when the boiler is fed with the distilled water from a surface condenser, and only the deficiency caused by the air-pump pumping out fresh water from the condenser in the form of aqueous vapor, is supplied by sea water, Mr. Belleville's apparatus would be injurious in removing the calcareous matter from the latter, for a thin deposit of scale on the heating surface of a marine boiler is absolutely necessary as a protection from corrosion on the water side. No *clean* iron in contact with sea water or diluted sea water can last long, and marine boilers always contain sea water, even when fed with distilled water from a surface condenser, because, although they may leave port filled with fresh water, the continuous loss of it by leakage and by the action of the air-pump pumping out vapor, and the substitution of an equivalent quantity of sea water, will soon exhaust the fresh water and replace it wholly by sea water.

For land boilers, however, the system of precipitating the salts of lime, from the feed water by high temperature is entirely beneficial, no evil resulting. It is a system that should be employed with all land boilers using feed water containing much calcareous matter. In France and Belgium, nearly all the fresh water contains a great deal of lime salts, and their removal from the feed water previous to its entering the boiler, is of the first consequence. The Belleville boiler, whose use is confined almost exclusively to those countries, was designed for their local conditions.

During the late war there were several large boilers in constant use in the naval station at Key West to provide potable water from the distillation of sea water for the United States fleet. Of course, these boilers being fed with sea water made a great deal of scale, to prevent which as much as possible the density of the boiler water had to be kept as low as practicable by "blowing off" supersalted water, which entailed a great loss of heat. Under these circumstances, Fleet Engineer Theodore Zeller, U. S. Navy, tried a system patented in England in 1852, by W. Danber, for taking the sulphate of lime out of the sea water before pumping the latter in the boiler. The system consisted in pumping the sea water first through a mixture of caustic barytes and oxalate of ammonia, which precipitated the lime, and then through filters which stopped the precipitate from entering the boiler. The system was perfectly successful so far as removing the sulphate of lime was concerned, but the corrosive action of the sea water on the clean iron of the boilers when unprotected by scale was so vigorous and rapid that it had to be abandoned.

In order to precipitate the lime from the feed water by heat alone, the latter must be exposed to a not less temperature than that due to saturated steam of about 105 pounds per square inch above the atmosphere; and as engines supplied with steam by Belleville boilers may have to work at a much less pressure, an automatic reducing valve has been attached to the boiler for the purpose of maintaining at least the 105 pounds pressure in it, while any desired less pressure may be used with the engine. When the initial pressure of the steam in the engine is not less than 105 pounds per square inch above the atmosphere, the reducing valve (which is merely an automatic throttle valve) is not to be used. The difference of pressure in the boiler and at the engine, caused by the reducing valve has both an advantage and a disadvantage.

The advantage, for either marine or land engines, consists in the fact that the steam room of the boiler is virtually increased in about the ratio of the absolute pressure on the opposite sides of the reducing valve, and priming or foaming correspond-

ingly lessened. Further, for land service, the precipitation of the calcareous matter from the feed water before pumping the latter into the boiler is wholly beneficial, but for marine service this precipitation is an evil, as it enables the sea water to powerfully corrode the iron of the boiler. Again, the reduction of the steam pressure at the reducing valve is at the expense of some of the heat in the steam, and is attended by a corresponding liquefaction of the latter. The higher pressure steam on one side of the reducing valve forces the same weight of steam, but with enlarged bulk, to the opposite side of the valve, and in so doing does mass work upon it measured by the product of the difference of the bulks of the steam on the two sides of the valve into the pressure on the engine or lower pressure side of the valve. For every 789¼ foot-pounds of work thus done there is consumed one Fahrenheit unit of heat. Further, the steam has a greater latent heat after the expansion than it had before the expansion, and the difference of these two latent heats has to be furnished from the heat of the steam, which, of course, undergoes an equivalent liquefaction to liberate sufficient heat to perform this molecular work. *Per contra*, there is available the difference between the heats represented by the product of the absolute temperature by the specific heat of saturated steam of the pressure before expansion, and the product of the absolute temperature by the specific heat of saturated steam of the pressure after expansion. The difference in the losses and gains of heat above described, due to the expansion of the steam at the reducing valve, is in the direction of loss. Some loss is always experienced there, which, being proportional to the difference of the pressure on the opposite sides of the valve, and as the Belleville system is operated generally, though not necessarily, with a reduction of pressure of from one-third to one-half of the boiler pressure, the waste of heat due to this expansion may be serious. The above waste is of the heat in the steam—that is, the heat of vaporization—but there is another important loss of heat due to the generation of steam under the higher pressure, namely, the increased temperature of the water from which it is generated. This quantity of heat, in the present case,

will be measured by the difference between the product of the temperature of the steam on one side of the reducing valve multiplied by the specific heat of water of that temperature, and the product of the temperature of the steam on the opposite side of the reducing valve multiplied by the specific heat of water of that temperature. When the quantity of heat thus obtained is added to the other losses, it will be found that any advantages obtained by the throttling of the steam at the reducing valve have been very dearly purchased. With a marine boiler supplied with distilled water from a surface condenser there is not the slightest necessity of a reducing valve. All the steam room necessary can be much more cheaply had by simply increasing the capacity of the steam drum. There may be added that the reducing valve is no part of the Belleville boiler proper, and that it can just as well be applied to any other boiler.

Obviously there remain against this system of using greatly throttled steam the important facts that the boiler must be made stronger to resist the greater pressure than what is used, and, consequently, heavier; that the losses of heat by radiation from the steam drum and feed pipe connecting it with the rectangularly cross-sectioned pipe collecting the feed water will be greater in the ratio of the difference between the atmospheric temperature and the temperatures of the steam before and after throttling, and that the gases of combustion will leave the boiler at a higher temperature the higher the temperature of the water and steam in it, involving more loss of heat. The latter two particulars affect the economy of the vaporization. There will be no loss in the Belleville boiler of heat radiated from the surfaces of the tubes and their connecting chambers, because they are immersed in the gases of combustion; but the losses of heat by radiation from the external casing and from the brick furnace walls may be very high on account of the high temperature of these gases on the inner side of the casing and walls, and the perviousness of the latter to heat, even when made of the most refractory materials. This remark applies to all "pipe" boilers, of which the Belleville is one variety, and accounts for some of the less economic vaporization given by them as against internally-fired boil-

ers having equally favorable proportions, and being used under equal conditions.

In the Belleville boiler the glass water gauge is so placed that when the water level is carried at midheight of the glass tube, as directed by Mr. Belleville, this level in the boiler is at the axis of the sixth row of tubes from the bottom, according to which level, *when the water is quiescent* in the boiler, five-eighths of the total heating surface would be water heating surface, and the remaining three-eighths would be steam superheating surface. This, however, is not Mr. Belleville's intention. He does not propose to superheat the steam at all. He supposes that the steam, in its passage through the tubes, will entrain or push along with it a sufficiently large volume of water to nearly "solidly" cover the *whole* of the total heating surface. In fact, he says that the quantity of water thus projected or carried along by the steam is, on the average, from four to six times greater than the quantity of feed water. He supposes that this enormous quantity of projected water will enter the bottom of the horizontal steam drum, together with the steam; that it will flow along the bottom of this drum, together with the feed water pumped into the drum, descend the pipe connecting the drum with the rectangularly sectioned pipe distributing the feed water to the elements, and thence again ascend the boiler tubes, and so on. In fact each element of the Belleville boiler, from bottom tube to top one, is actually filled, when the rate of combustion passes a certain low limit, with a mixture of steam and water. There is, under these conditions, no water level in the elements, although there may be a water level in the gauge. The weight of the quiescent water column in the gauge exactly counterbalances the weight of froth, foam or mixed steam and water in the elements, reckoning from their bottom tube to their top tube. It shows the level the water would be in the elements if the generation of steam was suddenly stopped. The bottom of the glass water gauge is connected with the rectangularly cross-sectioned pipe which collects the feed water, and the top of the gauge is connected with the sixth front chamber or box from the bottom.

The horizontal steam drum has a baffle plate extending nearly its entire length, and curving from bottom to top. The mingled mass of steam and water coming from the elements is projected against the bottom of this curved plate and supposed to be there separated, the steam rising over the top of the plate and the water remaining beneath it.

The feed water is pumped into one end of the horizontal steam drum against the pressure there, flows along the bottom of the drum to the other end, and then descends by gravity a vertical pipe connecting that end of the drum with the horizontal rectangularly cross-sectioned pipe collecting the feed water and distributing it to the tubes. The pressure at the top of this vertical pipe is, of course, the steam pressure in the drum; and the pressure in the bottom of the pipe is that pressure plus the pressure due to the height of the water column in the pipe. The pressure at the upper ends of the tubes discharging into the horizontal steam drum is, of course, the pressure of the steam there, and the pressure at the lower ends of the tubes receiving the feed water is the steam pressure plus the pressure due to the height of the water column in the tubes *when the boiler is not in operation*, but *when it is in operation* there must be added to the latter two pressures the pressure required to overcome the resistance of the wetted surfaces of the interior of the tubes and bends to the flow over them of the feed water and the water entrained by the steam, and to overcome the direct resistance of the semicircular bends connecting the tubes. The pressure at the lower ends of the tubes receiving the feed water will then be greater when the boiler is in operation than it was when the boiler was not in operation, by the pressure required to overcome all the resistances of the wetted surfaces and bends of the tubes to the flow of the feed water and of the steam entrained water, and this pressure must be equilibrated by an additional height of the water column in the vertical pipe sufficient to produce it. Hence, the water level in the glass water gauge will always be higher than what is due to the weight of water in the elements, by a height equivalent to the additional pressure described. With a slight feed the difference will be slight; as the feed increases the dif-

ference will increase in the proportion of the square of the quantity of the feed water and the entrained water entering the lower tube of the elements. After the combustion has reached a certain rate there will thus be no longer any head of water in the vertical pipe available to produce feeding, and, consequently, the maximum limit to the generation of steam will be reached, after which, if the rate of combustion be increased, the water in the tubes will diminish by vaporization faster than the feeding supplies, and will gradually leave the upper portion of the tube surface exposed to overheating, the water gauge continuing to indicate a higher water level in the elements than corresponds to the actual water in them.

Herein will be found the reason why the rate of combustion in the Belleville boiler cannot be forced beyond a very moderate limit—say 18 pounds of coal per hour per square foot of grate surface. It cannot be supplied with feed water for a greater rate of combustion, because the feeding depends on a pressure produced by a small head of water in a stand pipe, which head rapidly lessens as the rate of combustion (and quantity of water vaporized in a given time) increases, until the water head becomes nearly *nil.* If, nevertheless, the rate of combustion be increased beyond this limit the water will be nearly all vaporized from the tubes and they will be burned out, but any such great disappearance of the water from the tubes will be shown, of course, by the water gauge. If the feed pump throw in an increased quantity of water commensurate with the increased rate of combustion, that water will mostly accumulate in the horizontal steam drum until the latter is nearly filled and the engine stopped by the large mass of water thus passed into the steam pipe. Mr. Belleville provides against this contingency by means of an automatic regulator of the feed, but all experience shows that no such apparatus can be depended on in practice. Preventing the feed from exceeding a certain limit does not prevent the rate of combustion normal to that limit from being exceeded. The fall of the water in the gauge, if the automatic feed regulator be in proper operation, is the notice that the rate of combustion must be lessened.

The operation of the boiler, according to the intentions of its designer, requires that each element shall discharge such a volume of mingled steam and water into the horizontal steam drum with such force as to absolutely prevent any of the feed water pumped into that drum from entering the elements at the top, in which case evidently the boiler could not furnish any superheated steam; but, in fact, experiment has shown that with low rates of combustion the boiler does furnish such steam, from which there follows that, at least under the conditions of such experiments, the feeding was at the top of the elements instead of at the bottom. Indeed only the more or less rapid production of steam determines whether the boiler feeds at the top or at the bottom of its elements. At the higher rates of combustion practicable with it, there is no doubt that a sufficient mass of mixed steam and water is driven in a given time through the elements into the horizontal steam drum, to prevent the feed water pumped into the latter from entering the top tubes of the elements, while at lower rates of combustion only steam unmixed with water would be delivered into the drum, and would not be able to prevent the entrance of the feed water into the top tubes, in which case, if the rate of combustion be very slow, the steam would be superheated by the several upper rows of tubes which would then be nearly bare of water, the small quantity of feed water running along the lower sides of these tubes.

In experimenting with the Belleville boiler a pressure gauge should be applied to the rectangularly cross-sectioned pipe from which the lower front boxes or chambers are supplied with feed water, and a thermometer should be inserted in the steam drum. The pressure in that pipe and the temperature of the steam require to be known, in addition to the data usually taken in a boiler experiment.

From the foregoing will be seen that the feeding of the Belleville boiler is by gravity, and depends on the height of the water column in the vertical pipe connecting the steam drum and rectangularly cross-sectioned pipe lying across the front of the boiler, and distributing the feed water to the different elements. The height of this column is limited by the height of the boiler above

the lower tubes, and the feeding cannot be any faster than is due to the difference between the pressure in the rectangularly cross-sectioned pipe and the water pressure in the vertical pipe at the same level. The vertical pipe acts as a stand pipe, into which the feed pump deliivers, and the stand pipe feeds the boiler by gravity, so that the piston of the feed pump exercises no pressure on the feed water entering the elements. If more feeding is required, the difference between the two above described pressures must be made greater, and, after the vertical pipe is full, this can be only accomplished by lowering the water in the elements. The latter operation, however, uncovers the tubes and exposes them to destruction by the heat of the furnace.

Thus the rate of vaporization with this boiler is restricted, as above shown, and cannot be increased beyond what is due to the rate of combustion of, say, at the maximum, 18 pounds of bituminous coal per hour per square foot of grate. Of course, a mechanical draft forcing the rate of combustion above this limit cannot be employed.

The "nippling" of the connection of the lower front tube boxes or chambers with the rectangularly cross-sectioned pipe supplying them with feed water, whereby the aperture for the passage of the water from the latter to the former is much reduced, is in the direction of securing more uniformity of water distribution to the different elements, and is especially necessary when very low rates of combustion are used. The proper proportioning of these nipples is an extremely delicate matter, and they are essential to the safe working of the boiler, particularly under the condition just stated.

When the Belleville boiler is operated at so low a rate of combustion that its elements do not discharge water into the horizontal steam drum, but steam only, there is a distinct water level in each element, because neither the rate of vaporization nor the supply of feed water can scarcely ever be exactly the same for all. There is no knowing where any of these water levels are, and one or more elements may be overheated and destroyed, while others may have sufficiency of water. Of course, when

the water level in the elements is low enough the steam will be superheated.

As the elasticity (if such a term can be used in this connection) of the boiler is exceedingly restricted in the quantity of steam it can be made to generate in a given time, it is much better adapted for use when a regular supply is wanted than for naval use, in which the boiler must be able at times, by means of a mechanical draft, to double or treble the quantity of steam per hour due to its normal performance with natural draft.

The coil arrangement of the tubes securing the circulation of the water over every portion of the heating surface is all that could be desired, *provided a water level was maintained in the drum.*

What may be called the mechanical design of the boiler is very good. Its area on the floor is a minimum, being only that of its grate surface and furnace walls, a most valuable quality. The boxes or connecting chambers of the tubes being free to slide on each other, give all the necessary "play" or movement required for the elongation or contraction of the tubes by the great differences of temperature to which they are subjected. The openings (closed by plugs) in the front boxes allow the interior of these boxes and of the tubes to be inspected at will. The mean depth of each element over the tubes (45 inches) permit the tubes to be swept of soot in their places *when the boiler is cold.* Any element of the boiler, or group of tubes, can be withdrawn without disturbing any other part of the boiler, for examination, cleaning, repair or renewal. The joining of the tubes with their connecting boxes by screw threads allows the tubes and boxes to be separated easily with small hand tools and unskilled labor, *provided these joints do not rust fast, or the threads become distorted by the action of the high temperature of the gases of combustion surrounding them,* particularly in the case of the upper rows of tubes, which must often contain only highly superheated steam. But these purely mechanical advantages have been obtained at the expense of the economic vaporization of the boiler in an especial degree. The front of the boiler above the furnace is occupied wholly by doors to permit the examination and withdrawal of the elements as above described

Now as these doors cannot practically be made to close air-tight, there must always be an inleakage of the cold external air around their entire periphery, and this air mingling with the gases of combustion within the boiler, reduces their temperature and their heating efficiency. This air inleakage also injures the draft of the boiler by cooling the gases of combustion. The difference of the temperature of the inleaked air when entering the boiler and when leaving it represents so much heat abstracted from the gases of combustion.

The inleakage of air is something to be most carefully guarded against, and its prevention will warrant much sacrifice of mechanical advantages.

The boiler has been designed for regular land service in mills and factories; for consuming a uniform weight of coal per hour, furnishing a uniform supply of steam per hour, and maintaining a uniform pressure of steam; the rate of combustion per hour per square foot of heating surface not exceeding what can be given by the natural draft of ordinary heights of chimneys. It has several very ingenious automatic contrivances for maintaining constant the feed of the boiler, the pressure of the steam within it, the freeing of the feed water from calcareous matter, the separation of the steam from any entrained water accompanying it, etc. These devices are, however, no part of the boiler proper, and can be applied to other boilers equally well; and there may be said of them in general that, like all automatic contrivances applied to any boiler, they cannot be trusted, so after all, the real regulation has to be performed by human hands directed by human intelligence and character. They may, indeed, assist the intelligence by rendering its labor less, but will in the end betray it, if confided in.

In the naval service irregularity, and not regularity, is the rule. The power required is continually varying. The steam pressure, consumption of coal, speed of engine, and all conditions of the machinery are in constant change, so that here especially no automatic mechanism can judiciously be used, besides which the sudden and violent movements of sea steamers by wind and wave almost wholly precludes the operation of such devices.

Under these conditions, automatic contrivances are "traps" in more senses than the mechanical one.

The following are the principal dimensions and proportions of the boiler:

Outside breadth of boiler proper	5 feet 2 inches.
Outside length of boiler proper	6 feet 10½ inches.
Height of boiler to top of steam drum	10 feet 3½ inches.
Breadth of grate surface	3 feet 4 inches.
Length of grate surface	4 feet 9.6 inches.
Area of grate surface	16 square feet.
Diameter of chimney	2 feet.
Height of chimney above level of grate	46 feet.
Cross area of chimney	3.1416 square feet.
Number of tubes (iron)	64.
Outside diameter of tubes	3.937 inches.
Inside diameter of tubes	3.562 inches.
Length of tubes in clear of boxes or bends	5 feet 2 inches.
Calorimeter of the tubes, or aggregate area between them and between them and the sides	3.661444 square feet.
Calorimeter of the tubes after diminution by baffle plates	2.288402 square feet.
Outside heating surface in the rectangularly cross-sectioned pipe	4.16 square feet.
Inside heating surface in the rectangularly cross-sectioned pipe	3.333 square feet.
Outside heating surface in the four lower front chambers	10.40 square feet.
Inside heating surface in the lower front chambers	8.00 square feet.
Outside heating surface in the twenty-four front chambers, above the lower four	32.40 square feet.
Inside heating surface in the twenty-four front chambers above the lower four	22.08 square feet.
Outside heating surface in the thirty-two back chambers	48.864 square feet.
Inside heating surface in the thirty-two back chambers	39.104 square feet.
Outside heating surface in the four upper bends	4.944 square feet.
Inside heating surface in the four upper bends	3.112 square feet.
Heating surface in the sixty-four tubes, calculated for their outside diameter	340.82 square feet.
Heating surface in the sixty-four tubes, calculated for their inside diameter	308.36 square feet.
Total heating surface with the surface of the tubes and connecting boxes calculated for their outside dimensions	441.588 square feet.
Total heating surface with the surface of the tubes and connecting boxes calculated for their inside dimensions	383.999 square feet.
Ratio of the total heating surface with the surface of the tubes and connecting boxes calculated for their outside dimensions, to the grate surface	27.599
Ratio of the total heating surface with the surface of the tubes and connecting boxes calculated for their inside dimensions, to the grate surface	24.000
Ratio of the grate surface to the cross-section of the chimney	5.093

Ratio of the grate surface to the calorimeter between the tubes..... 4.370
Ratio of the grate surface to the calorimeter actually used (after diminution of the spaces between tubes by baffle plates)..... 7.000

Whatever mechanical advantages the Belleville boiler may have, can be preserved and the boiler adapted for naval use according to the requirements of the present day, by making a few modifications—omissions, substitutions and additions.

In the first place, the various automatic mechanisms, the steam-reducing valve, etc., must be removed, and the system for precipitating the calcareous matter in the sea water and for making good the fresh water lost by the action of the air pump in pumping out vapor, and by other causes, must be dispensed with. In the next place, for the vertical standpipe connecting the steam drum and the rectangularly cross-sectioned pipe distributing the feed water to the elements, and the vessel for receiving the lime salts precipitate, there must be substituted a high cylindrical "separator" for separating, by gravity, the steam from its entrained water. This separator is to communicate with the steam drum at bottom and at top of each for water and steam, and with an auxiliary feed pump or circulating pump at the bottom. The office of this auxiliary feed pump or circulating pump is to draw the entrained water from the bottom of the separator, together with whatever quantity of water additional to the feed water pumped into the boiler by the regular feed pump, may be circulated through the boiler in order to have the *total* heating surface made water-vaporizing surface *always*, and to force these differently derived waters back into the boiler.

The circulation through the boiler of a quantity of water additional to the feed water, and the maintaining of a water level in the horizontal steam drum a little above the mouths of the tubes opening into it, are indispensable to the safe, easy and economical working of any boiler of this type, at sea, under the conditions of modern naval use.

This additional or circulating water should, in quantity be at least equal to several times the feed water, and is kept circulating through the boiler tubes by the special circulating pump, just described, for the purpose stated, which pump also draws

the entrained water from the "separator," and forces it, together with the circulating water, into the boiler. The only power expended in this pumping is what is required to overcome the various frictions of the mechanisms and the difference between the pressures in the "separator" and in the rectangularly cross-sectioned pipe.

The feed water proper, instead of being pumped into the steam drum, as at present, should be pumped directly into the rectangularly cross-sectioned pipe by the regular feed pump.

The doors opening into the space containing the tubes should be closed by screws around the whole periphery and pressing against air-tight packing. These doors to be opened only at considerable intervals.

The arrangement above recommended enables whatever quantity of water, feed or circulating, that may be required, to be supplied mechanically to the boiler. It enables the whole of the heating surface to be made efficient water vaporizing surface and thereby increases the economic efficiency. It water-sweeps the whole of that surface with great velocity, sweeping off the steam bubbles as fast as generated, and it causes a thorough and rapid mechanical or forced circulation of the water throughout the whole of the boiler, not to be otherwise accomplished, besides securing a uniform distribution of the water to the different elements under all circumstances.

With this system as much coal could be burned on the square foot of grate per hour in this boiler as in any other, and mechanical or forced draft could be used with it to the maximum extent.

The top of the "separator" is, of course, steam room, and to it should be attached the steam pipe to the engine. Alterations of the present proportions in several ways, would be judicious in combination with the system recommended. For instance, the heating surface should be increased by the addition of at least two rows of tubes in height (to each element), and, preferably, four rows. This addition increases the height of the boiler from one to two feet; but even in the latter case not too high for use in any large naval vessel, and still remain below the level of the sea.

The large diameter of the tubes (nearly 4 inches) requires, of course, for equal pressure, greater thickness of metal than the much smaller diameter tubes of most other pipe boilers. The heating surface, therefore, of the Belleville pipe boiler will be thicker, and to that extent, for equal areas, heavier than the heating surface of most other pipe boilers, but its durability will be proportionally greater. The metal of the other parts of the Belleville boiler need not be any thicker than is used with any other pipe boiler, the durability being equal. If the tubes are to be made removable for examination, cleaning and repair, their larger diameter, greater thickness of metal and fewer number, are real practical recommendations of importance.

The writer is of opinion that with the alterations he has recommended, this pipe boiler, skilfully detailed, would possess qualities for naval use in large vessels, and using mechanical draft to the fullest extent desirable, second to no other pipe boiler.

EXPERIMENTS.

In the following table, No. 1, are contained the observed data and the calculated results of three experiments, the last two being repetitions of the first, made on three different days with the Ericsson engine and the Belleville boiler. In these experiments the differences of conditions were so slight that the three have been considered as one, with the advantage that the results from the aggregate are more accurate than from either taken alone.

For facility of reference the quantities have been grouped, and each is so fully described in the table that only a few need further explanation.

The engine was a non-condensing one, and ought to have been furnished with a feed-water heater, whereby the temperature of that water might have been made about 212 degrees Fahrenheit, instead of only the 83 degrees that it was. Now, the temperature of the feed water greatly affects the economic performance of the engine, whether estimated by weight of fuel consumed or by number of units of heat consumed per hour per horse power developed by the engine; and, as the 83 degrees

temperature of the feed water was purely an accidental one, and not connected with the necessary performance of the engine, a proper determination of the economy of that performance requires it to be ascertained for the temperature of feed water (212 degrees) necessarily connected with the engine, as the highest attainable, when using steam without condensation. Of course, the temperature of the feed water has no influence upon the economy of the performance when the cost of the horse power is estimated in pounds of feed water consumed per hour. In the table, therefore, the cost of the indicated, net and total horse powers is given for the pounds of crude coal, for the pounds of the combustible portion of that coal, and for the number of Fahrenheit units of heat consumed per hour, on the supposition that the temperature of the feed water was 212 degrees, the actual weights of coal and of combustible consumed per hour having been modified in the ratio required by the difference between the feed-water temperatures, 83 and 212 degrees. For the weight of feed water consumed per hour per horse power no modification was needed. The number of Fahrenheit units of heat consumed per hour during the experiment is the number of pounds of feed water consumed during that time multiplied by the difference between the number of Fahrenheit units in one pound of water at the temperature of 212 degrees, and in one pound of saturated steam at the temperature (406 degrees) of the steam in the boiler.

In calculating the weight of steam liquefied in the cylinders to liberate the heat transmuted into the work done by the expanding steam, the mechanical equivalent of one Fahrenheit unit of heat is taken at $789\frac{1}{4}$ pounds.

In calculating the weight of steam present in the cylinders, allowance has been made for the weight already in the cylinders at the commencement of the stroke of the pistons due to the cushioned back pressure.

TABLE NO. 1.

CONTAINING THE MEAN DATA AND RESULTS OF THREE REPETITIONAL EXPERIMENTS MADE IN 1889 ON THE HORIZONTAL, NON-CONDENSING, SINGLE-ACTING, COMPOUND STEAM ENGINE WITHOUT STEAM JACKETS, DESIGNED BY JOHN ERICSSON AND USED IN THE "DELAMATER IRON WORKS" OF NEW YORK CITY.

The same conditions obtained during all the experiments. The first experiment was made on the 28th of March, and continued 9 hours and 10 minutes consecutively; the second was made on the 3d of April, and continued 11½ consecutive hours; and the third was made on the 4th of April, and continued 11 consecutive hours. A complete set of indicator diagrams was taken every half hour. No cut-off valves on the large cylinder.

The steam was supplied by a Belleville boiler.

TOTAL QUANTITIES.

Number of sets of indicator diagrams, one from the acting end of each cylinder to a set, from which the following mean cylinder pressures were taken	63.
Aggregate duration of the experiments in hours and minutes	31.40
Total number of pounds of feed water vaporized	31,637.4
Total number of double strokes made by the pistons of the engine,	284,050.
Total number of pounds of ordinary semi-bituminons coal consumed	4,444.
Total number of pounds of refuse from the coal, in ash, clinker, and unconsumed pieces of coal fallen through the grate	454.
Total number of pounds of combustible consumed, or coal less the refuse	3,990.
Per centum of coal in refuse of ash, clinker and unconsumed pieces	10.216022

ENGINE.

Steam pressure in boiler in pounds per square inch above the atmosphere	249.8
Steam pressure in boiler in pounds per square inch above zero	264.5
Steam pressure in the steam pipe after the steam had passed through the reducing valve, in pounds per square inch above the atmosphere	199.8
Steam pressure in the steam pipe after the steam had passed through the reducing valve, in pounds per square inch above zero	214.5
Difference between the steam pressure in the boiler and the steam pressure in the steam pipe after the steam had passed through the reducing valve, in pounds per square inch	50.0
Position of throttle valve	Wide open.
Number of double strokes made per minute by the pistons of the engine	149.5

Fraction completed of the stroke of the piston of the small cylinder when the steam was cut off	0.392
Fraction completed of the stroke of the piston of the small cylinder when the steam was released	0.975
Fraction completed of the stroke of the piston of the small cylinder when the steam was cushioned	0.975
Fraction completed of the stroke of the piston of the large cylinder when the steam was released	0.975
Fraction completed of the stroke of the piston of the large cylinder when the steam was cushioned	0.282
Number of times the steam was expanded in the small cylinder	2.239225
Number of times the steam was expanded in the engine	8.808361
Speed of the pistons per minute in feet	398⅝
Number of pounds of feed water consumed per hour	999.096823
Number of pounds of coal that would have been consumed per hour, had the temperature of the feed water been 212 degrees Fahrenheit	124.553323
Number of pounds of combustible that would have been consumed per hour, had the temperature of the feed water been 212 degrees Fahrenheit	111.828928
Number of Fahrenheit units of heat consumed per hour above the temperature 212 degrees Fahrenheit of feed water	1,023,902.833630

TEMPERATURES.

Temperature in degrees Fahrenheit of the steam in the boiler	406.
Temperature in degrees Fahrenheit of the steam in the steam pipe after the steam had passed through the reducing valve	387.68
Temperature of the gases of combustion in the chimney of the boiler	Antimony melted.
Difference between the temperature of the steam in the boiler and the temperature of the steam in the steam pipe after the steam had passed through the reducing valve	18.32
Temperature in degrees Fahrenheit of the feed water entering the boiler	83.

RATE OF COMBUSTION.

Number of pounds of coal consumed per hour	140.336842
Number of pounds of combustible consumed per hour	126.
Number of pounds of coal consumed per hour per square foot of grate surface	8.771053
Number of pounds of combustible consumed per hour per square foot of grate surface	7.875000
Fraction of a pound of coal consumed per hour per square foot of inner total heating surface	0.365462
Fraction of a pound of combustible consumed per hour per square foot of inner total heating surface	0.328126

STEAM PRESSURES IN SMALL CYLINDER PER INDICATOR.

Pressure on the piston of the small cylinder at the commencement of its stroke, in pounds per square inch above zero	196.83
Pressure on the piston of the small cylinder at the point of cutting off the steam, in pounds per square inch above zero	152.
Pressure on the piston of the small cylinder at the end of its stroke, in pounds per square inch above zero	68.23
Mean back pressure against the piston of the small cylinder, in pounds per square inch above zero, inclusive of the cushioning	30.1
Mean back pressure against the piston of the small cylinder, in pounds per square inch above zero, exclusive of the cushioning	30.05
Back pressure against the piston of the small cylinder at the point in its stroke where the cushioning began, in pounds per square inch above zero	21.
Indicated pressure on the piston of the small cylinder, in pounds per square inch	98.615
Net pressure on the piston of the small cylinder, in pounds per square inch	94.615
Total pressure on the piston of the small cylinder, in pounds per square inch above zero	127.914
Total pressure on the piston of the small cylinder of the expanding steam alone, in pounds per square inch above zero	105.

STEAM PRESSURES IN LARGE CYLINDER PER INDICATOR.

Pressure on the piston of the large cylinder at the commencement of its stroke, in pounds per square inch above zero	45.98
Pressure on the piston of the large cylinder at the end of its stroke, in pounds per square inch above zero	18.47
Mean back pressure against the piston of the large cylinder, in pounds per square inch above zero, inclusive of the cushioning	17.54
Mean back pressure against the piston of the large cylinder, in pounds per square inch above zero, exclusive of the cushioning	15.55
Back pressure against the piston of the large cylinder at the point in its stroke where the cushioning began, in pounds per square inch above zero	16.
Indicated pressure on the piston of the large cylinder, in pounds per square inch	9.144
Net pressure on the piston of the large cylinder, in pounds per square inch	8.144
Total pressure on the piston of the large cylinder, in pounds per square inch above zero	20.309
Total pressure on the piston of the large cylinder of the expanding steam, in pounds per square inch above zero	26.684

EQUIVALENT INDICATED AND NET PRESSURES.

Indicated pressure in pounds per square inch that would be on the piston of the large cylinder were the indicated pressure on the piston of the small cylinder divided by the ratio (3.933665) of the area of the small to the area of the large piston and the quotient added to the experimental indicated pressure on the piston of the large cylinder	34.2135
Net pressure in pounds per square inch that would be on the piston of the large cylinder were the net pressure on the piston of the small cylinder divided by the ratio (3.933665) of the area of the small to the area of the large piston and the quotent added to the experimental net pressure on the piston of the large cylinder	32.1966

HORSES-POWER.

Indicated horses-power developed in the small cylinder	29.971120
Net horses-power developed in the small cylinder	28.755438
Total horses-power developed in the small cylinder	38.875687
Horses-power developed in the small cylinder by the expanding steam alone	19.402285
Indicated horses-power developed in the large cylinder	10.931850
Net horses-power developed in the large cylinder	9.736328
Total horses-power developed in the large cylinder	18.107527
Horses-power developed in the large cylinder by the expanding steam alone	23.791484
Aggregate indicated horses-power developed by the engine	40.902970
Aggregate net horses-power developed by the engine	38.491766
Aggregate total horses-power developed by the engine	56.983214
Aggregate horses-power developed by the expanding steam alone in both cylinders	43.193769
Per centum which the aggregate indicated horses-power are of the aggregate total horses-power	71.780735
Per centum which the aggregate net horses-power are of the aggregate total horses-power	67.549307
Per centum which the aggregate horses-power required to overcome the back pressure against the pistons are of the aggregate total horses power	28.219267

WEIGHT OF STEAM ACCOUNTED FOR BY THE INDICATOR.

Number of pounds of steam present per hour in the small cylinder at the point of cutting off the steam, calculated from the pressure there	663.188404
Number of pounds of steam present per hour in the small cylinder at the end of the stroke of its piston, calculated from the pressure there	726.070378

Number of pounds of steam condensed per hour in the small cylinder to furnish the heat transmuted into the total horses-power developed by the expanding steam alone in that cylinder	55.294917
Sum of the two immediately preceding quantities	781.365295
Number of pounds of steam present per hour in the large cylinder at the end of the stroke of its piston, calculated from the pressure there	595.439510
Number of pounds of steam condensed per hour in the small and large cylinders to furnish the heat transmuted into the total horses-power developed by the expanding steam alone in those cylinders after the closing of the cut-off valve on the small cylinder	118.574049
Sum of the two immediately preceding quantities	714.013559

DIFFERENCE BETWEEN THE WEIGHT OF STEAM EVAPORATED IN THE BOILER AND THE WEIGHT OF STEAM ACCOUNTED FOR BY THE INDICATOR.

Difference, in pounds per hour, between the weight of steam evaporated in the boiler and the weight of steam accounted for by the indicator at the point of cutting off the steam in the small cylinder	335.908419
Difference, in per centum of the weight of steam evaporated in the boiler, between that weight and the weight of steam accounted for by the indicator at the point of cutting off the steam in the small cylinder	33.621208
Difference, in pounds per hour, between the weight of steam evaporated in the boiler and the weight of steam accounted for by the indicator at the end of the stroke of the piston of the small cylinder	217.731528
Difference, in per centum of the weight of steam evaporated in the boiler, between that weight and the weight of steam accounted for by the indicator at the end of the stroke of the piston of the small cylinder	21.792836
Difference, in pounds per hour, between the weight of steam evaporated in the boiler and the weight of steam accounted for by the indicator at the end of the stroke of the piston of the large cylinder	285.083264
Difference, in per centum of the weight of steam evaporated in the boiler, between that weight and the weight of steam accounted for by the indicator at the end of the stroke of the piston of the large cylinder	28.534098

ECONOMIC RESULTS—TEMPERATURE OF FEED WATER 212° FAHRENHEIT.

Number of pounds of feed water consumed per hour per indicated horse-power developed by the engine	24.426021
Number of pounds of feed water consumed per hour per net horse-power developed by the engine	25.956118

Number of pounds of feed water consumed per hour per total horse-power developed by the engine	17.533178
Number of Fahrenheit units of heat above the temperature 212 Fahrenheit degrees, consumed per hour per indicated horse-power developed by the engine	25,032.481350
Number of Fahrenheit units of heat above the temperature 212 Fahrenheit degrees, consumed per hour per net horse-power developed by the engine	26,600.567862
Number of Fahrenheit units of heat above the temperature 212 Fahrenheit degrees, consumed per hour per total horse-power developed by the engine	17,968.499173
Number of pounds of coal consumed per hour per indicated horse-power developed by the engine	3.045092
Number of pounds of coal consumed per hour per net horse-power developed by the engine	3.235843
Number of pounds of coal consumed per hour per total horse-power developed by the engine	2.185790
Number of pounds of combustible consumed per hour per indicated horse-power developed by the engine	2.734005
Number of pounds of combustible consumed per hour per net horse-power developed by the engine	2.905269
Number of pounds of combustible consumed per hour per total horse-power developed by the engine	1.962489

VAPORIZATION.

Number of pounds of water that would have been vaporized in the boiler had it been supplied at the temperature of 100 degrees Fahrenheit and vaporized under the standard atmospheric pressure	33,872.246042
Number of pounds of water that would have been vaporized in the boiler had it been supplied at the temperature of 212 degrees Fahrenheit and vaporized under the standard atmospheric pressure	37,829.904244
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 100 degrees Fahrenheit by one pound of coal	7.622018
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 100 degrees Fahrenheit by one pound of combustible	8.489285
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 212 degrees Fahrenheit by one pound of coal	8.512580
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 212 degrees Fahrenheit by one pound of combustible	9.481180
Number of pounds of water vaporized per hour under the atmospheric pressure from the temperature of 212 degrees Fahrenheit by one square foot of inner total heating surface	3.111019

In the following table No. 2, will be found the performance of the Belleville boiler alone during two experiments of ten consecutive hours each—one made on the 30th of November and the other made on the 5th of December, 1888. The steam in this case was used by one of the shop engines of the Delamater Iron Works, and, as the circumstances of the two experiments varied in but a very slight degree, the aggregate of both has been considered as one experiment. The only determination made by these experiments was the economic efficiency of the boiler under the experimental conditions.

The experiments were made by the same persons using the same instruments, tanks, scales, etc., as for the experiments just detailed in Table No. 1. The same kind of semi-bituminous coal was also used, and was fired by the same fireman.

TABLE NO. 2.

CONTAINING THE MEAN DATA AND RESULTS OF TWO REPETITIONAL EXPERIMENTS MADE IN 1888 ON THE EXPERIMENTAL BELLEVILLE BOILER IN THE "DELAMATER IRON WORKS" OF NEW YORK CITY.

The same conditions obtained during both experiments, the first of which was made on the 30th of November and the last of which was made on the 5th of December, both continuing ten consecutive hours.

TOTAL QUANTITIES.	
Aggregate duration of the experiments in hours	20.
Total number of pounds of feed water vaporized	23,971.4
Total number of pounds of ordinary semi-bituminous coal consumed	2,944.5
Total number of pounds of refuse from the coal, in ash, clinker, and unconsumed pieces of coal fallen through the grate	409.5
Total number of pounds of combustible consumed, or coal less the refuse	2,535.
Per centum of coal in refuse of ash, clinker and unconsumed pieces	13.907285
STEAM PRESSURES.	
Steam pressure in boiler in pounds per square inch above atmosphere	165.3
Steam pressure in boiler in pounds per square inch above zero	180.
Steam pressure in the steam pipe after the steam had passed through the reducing valve, in pounds per square inch above the atmosphere	86.3

Steam pressure in the steam pipe after the steam had passed through the reducing valve, in pounds per square inch above zero	101.
Difference between the steam pressure in the boiler and the steam pressure in the steam pipe after the steam had passed through the reducing valve, in pounds per square inch	79.
Number of pounds of feed water consumed per hour	1,198.57
TEMPERATURES.	
Temperature in degrees Fahrenheit of the steam in the boiler	372.97
Temperature in degrees Fahrenheit of the steam in the steam pipe after the steam had passed through the reducing valve	328.30
Temperature of the gases of combustion in the chimney of the boiler	Lead barely melted.
Difference in Fahrenheit degrees between the temperature of the steam in the boiler and the temperature of the steam in the steam pipe after the steam had passed through the reducing valve	44.67
Temperature in degrees Fahrenheit of the feedwater entering the boiler	131.
RATE OF COMBUSTION.	
Number of pounds of coal consumed per hour	147.225
Number of pounds of combustible consumed per hour	126.750
Number of pounds of coal consumed per hour per square foot of grate surface	9.201562
Number of pounds of combustible consumed per hour per square foot of grate surface	7.921875
Fraction of a pound of coal consumed per hour per square foot of inner total heating surface	0.383399
Fraction of a pound of combustible consumed per hour per square foot of inner total heating surface	0.330079
VAPORIZATION.	
Number of pounds of water that would have been vaporized in the boiler had it been supplied at the temperature of 100 degrees Fahrenheit and vaporized under the standard atmospheric pressure	24,371.122219
Number of pounds of water that would have been vaporized in the boiler had it been supplied at the temperature of 212 degrees Fahrenheit and vaporized under the standard atmospheric pressure	27,218.662107
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 100 degrees by one pound of coal	8.276829
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 100 degrees by one pound of combustible	9.613855

Number of pounds of water vaporized under the atmospheric pressure from the temperature of 212 degrees by one pound of coal	9.243900
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 212 degrees by one pound of combustible	10.737145
Number of pounds of water vaporized per hour under the atmospheric pressure from the temperature of 212 degrees by one square foot of inner total heating surface	3 544105

REMARKS.

As regards the engine, there was nothing either original or remarkable in the design, the only feature patented, namely, the combination with the engine of a check valve formed of a disc with conical sides resting in a corresponding conical seat, and placed at the top of the small cylinder in the passage leading to the condenser or to the atmosphere, so as to constantly keep almost an absolute vacuum above the piston of the small cylinder, was not used during the experiment, this poppet valve having been removed.

What may be indeed considered as remarkable in the original patent was that a similar check valve was not applied to the similar passage from the bottom of the large cylinder leading to the condenser or to the atmosphere; for evidently, whatever may have been the purpose of this valve (and the patentee gives no information on that point), as applied to the small cylinder, the same conditions existed requiring its application to the large cylinder. Although the patentee gave no explanation of the purpose to be answered by this check valve at one end of the small cylinder, nor any description of its mode of action, he evidently thought that by its means one side of the piston of the small cylinder could be relieved of opposing back pressure during the steam stroke (which is quite true), and that a corresponding economic gain could be made (which is quite false). The engine seems to have been originally built to practically test this preposterous question of economy with and without the valve in use, which certainly required no testing. Not the slightest difference of effect in relation to either pressure or economy attended the use or non-use of the valve. Of course, the use of

such a valve required the cylinders to be made single acting, with the attendant unquestionable disadvantage that, for equal development of power under equal conditions otherwise, the capacities of the cylinders had to be doubled, involving doubling the weight, bulk and cost of the engine. The obvious and elementary fact seems to have been quite overlooked that, with the single-acting arrangement of the cylinders, exactly whatever existing pressure opposed the movement of the pistons during one stroke, assisted the movement of the pistons during the return stroke, so that neither loss nor gain was possible.

The value of the experiment with the engine as built consists in the reliable determination of the cost of the horse power developed by it under the experimental conditions, in pounds of feed water consumed per hour, in Fahrenheit units of heat consumed per hour, and in pounds of crude coal and of the combustible portion of that coal consumed per hour.

The pressures of expanding steam in a steam engine, as given by the law of Mariotte, admit of very easy calculation, and as a knowledge of these pressures is a great convenience in steam engineering, it is of importance to ascertain how nearly they approximate to the pressures obtained from the indicator diagrams taken from a practical steam engine. For this purpose, the pressures of the expanding steam in the small cylinder of the experimental engine has been taken for different ordinates of all the diagrams, and the pressures according to the Mariotte law have been computed for the same ordinates. The results are given in the following table. The diagrams were large, smooth and accurate, having been taken by a fine instrument. The conditions were: Fraction of the stroke of the piston of the small cylinder when the steam was cut off, 0.392; fraction which the steam space in the passage and clearance at one end of the small cylinder is of the space displacement of the piston of that cylinder per stroke, 0.0986292; steam pressure in the small cylinder at the point of cutting off the steam, 152 pounds per square inch above zero. The back-pressure steam was cushioned or compressed through only 2½ per centum of

the stroke of the piston, and the piston (single-acting) made 149.5 double strokes per minute. The cylinder was not steam jacketed. The mean back pressure against the piston was 30.1 pounds per square inch above zero.

Fraction of the stroke of the piston of the small cylinder at which the ordinate of the indicator diagram is taken.	Steam pressure in pounds per square inch above zero, according to the ordinates of the indicator diagrams.	Steam pressure in pounds per square inch above zero, according to the Mariotte law, at the ordinates.
0.392	152.000	152.000
0.45	132.875	135.931
0.55	111.367	114.974
0.65	96.677	99.616
0.75	85.220	87.878
0.85	76.779	78.614
0.95	70.104	71.117
End of stroke.	68.230	67.881

The above table shows that the curve of the expanding steam, according to the indicator diagrams, is a little below (about 2¼ per centum below), but parallel to the curve, according to the Mariotte law. When the possible errors of data are considered, such as the space in the clearance and steam passage at one end of the cylinder, the spring of the indicator, the exact point at which the steam was cut off, etc., all or any of which might be equal to the difference between the curves, the conclusion to be drawn is that the experimental curve of expanding steam in a working cylinder was identical with the Mariotte curve—a mere accidental coincidence, but having convenient results.

As regards the performance of the Belleville boiler, there will be observed that the rate of combustion was almost the same in the two experiments (Tables Nos. 1 and 2), and quite low—about 9 pounds of coal per hour per square foot of grate surface—notwithstanding that semi-bituminous coal was burned, and that the temperature with which the gases of combustion left the boiler was high (antimony melted in one experiment, and lead melted in the other experiment, in the bottom of the chimney).

The height of the chimney—46 feet above the level of the grate—was quite low also. The low rate of combustion should be conducive to economic vaporization, nevertheless the vaporisative result was very mediocre in that respect.

In computing the economic vaporization in the two experiments of Tables Nos. 1 and 2, proper corrections must be made for the effect of the different temperatures due to the different steam pressures in the boiler, and in doing this, the temperature of the gases of combustion in the furnace at the moment of their generation will be taken at 1,800 degrees Fahrenheit.

The economic vaporizations given in the tables as under the atmospheric pressure from the temperature of feed water of, say, 212 degrees Fahrenheit, is really from that temperature under the experimental pressures, and not under the atmospheric pressure. Had they been under the latter, they would have been greater by the per centum that the difference between the temperatures of 212° and the experimental temperature of the boiler steam is of 1,800°.

Consequently, in Table No. 1, the temperature of the boiler steam being 406 degrees Fahrenheit, the number of pounds of water vaporized from the temperature of 212° under the standard atmospheric pressure by one pound of crude coal, will be

$$\left(\tfrac{406-212}{1,800}+1\right)\times 8.512580 = 9.425299 \text{ pounds.}$$

And by one pound of the combustible portion of that coal

$$\left(\tfrac{406-212}{1,800}+1\right)\times 9.481180 = 10.497752 \text{ pounds.}$$

In Table No. 2, the temperature of the boiler steam being 372.97 degrees Fahrenheit, the number of pounds of water vaporised from the temperature of 212° under the standard atmospheric pressure by one pound of crude coal will be

$$\left(\tfrac{372.97-212}{1,800}+1\right)\times 9.24390 = 10.070563 \text{ pounds.}$$

And by one pound of the combustible portion of that coal

$$\left(\tfrac{372.97-212}{1,800}+1\right)\times 10.737145 = 11.697346 \text{ pounds.}$$

The difference in the economic vaporizations in Tables Nos. 1 and 2, in pounds of water vaporized from the temperature of

212 degrees Fahrenheit and under the atmospheric pressure by one pound of the combustible portion of the coal, is $\frac{11.697346 - 10.497762 \times 100}{11.697346} = 10.2553$ per centum of the quantity in Table No. 2.

The difference between the temperatures of the gases of combustion in the chimney in the two experiments (melting points of lead and antimony,) is about 200 degrees Fahrenheit; consequently, the economic vaporization in the experiment of Table No. 1 should have been about $(\frac{200 \times 100}{1,800} =)$ 11.1111 per centum less than in the experiment of Table No. 2, thus agreeing very closely with the determination (10.2553 per centum) in the immediately preceding paragraph.

Experiment on the Belleville Boiler in the Steam Yacht *Shearwater*.

As a sequel to the foregoing, there may be here added the data and results of a trial made in November, 1887, of the economic vaporization of the Belleville boiler on board the steam yacht *Shearwater*, belonging to Mr. J. M. Forbes. This boiler was furnished direct from the Belleville Works, at St. Denis, France, and has exactly double the frontage and double the number of elements of the Belleville boiler experimented with at the Delamater Iron Works of New York city, but in all other respects the two are duplicates.

During the trial the steam was worked off by the engine, while the vessel remained attached to the dock at Wood's Holl, Massachusetts.

The trial was most carefully made by a Board of United States Naval Engineers, of which Chief Engineer A. S. Greene, U. S. N., was the presiding officer; and from his report to the Chief of the Bureau of Steam Engineering, Navy Department, printed in the

annual report of the Chief of that Bureau for 1888, the following data has been taken:

The grate surface of the boiler was 32 square feet, and the total heating surface 767.998 square feet, with the tubes and connections measured by their inside dimensions. The chimney was 29 feet high above the level of the grate, and 31 inches in diameter.

The experiment was begun and ended with the fire in full operation, and of the same thickness and cleanness as well as could be judged. The water level in the glass water gauge was also the same at the beginning and end of the trial. The fuel consumed was semi-bituminous coal from the Cumberland mines of Maryland, not selected and of average quality. All the coal consumed was accurately weighed on tested scales, and the ash, clinker and dust from it were also weighed upon the same scales and in the dry state. The feed water, previous to being pumped into the boiler, was measured in two wooden tanks, the water contents of which were ascertained by actual weighing. A mercurial thermometer was inserted in the stop-valve chamber of the steam drum, and another was inserted in the throttle-valve chamber near the engine. Thermometers were also placed in the measuring tanks, on deck in the shade, and in the fire room. The height of the barometer was regularly recorded. The temperature of the gases of combustion in the uptake of the boiler just above the tubes was approximately ascertained by placing there small pieces of tin, lead and zinc at back and front of boiler. No steam was admitted to the furnace over the incandescent coal, nor was the fire forced in any way. The sharp edges of the zinc were found after the trial to be slightly fused, the lead was completely melted and partially oxidized, and the tin had wholly disappeared.

The vessel was fitted with two permanent tanks containing in the aggregate 14,000 pounds of water for supplying the water waste of the boiler.

A complete set of observations was taken hourly and entered in the appropriate columns of a log, the totals and means of which are as follows:

TOTAL QUANTITIES.

Duration of the experiment in consecutive hours	24.
Total number of pounds of feed water vaporized	85,570.
Total number of pounds of semi-bituminous coal consumed	10,500.
Total number of pounds of refuse from the coal in ash, clinker and unconsumed pieces of coal fallen through the grate	808.
Total number of pounds of combustible consumed, or coal less the refuse	9,692.
Per centum of coal in refuse of ash, clinker and unconsumed pieces	7.695238

STEAM PRESSURES.

Height of the barometer in inches of mercury	30.33
Pressure of the atmosphere, in pounds per square inch above zero	14 888
Steam pressure in boiler, in pounds per square inch above atmosphere	111.34
Steam pressure in boiler, in pounds per square inch above zero,	126.228
Steam pressure in the steam pipe after the steam had passed through the reducing valve, in pounds per square inch above the atmosphere	88.4
Steam pressure in the steam pipe after the steam had passed through the reducing valve, in pounds per square inch above zero	103.288
Difference between the steam pressure in the boiler and the steam pressure in the steam pipe after the steam had passed through the reducing valve, in pounds per square inch	22.94
Number of pounds of feed water consumed per hour	3,565.416667

TEMPERATURES.

Temperature in degrees Fahrenheit of the steam in the boiler, by thermometer	353.42
Temperature in degrees Fahrenheit of the steam in the boiler considered as saturated	344 87
Number of degrees Fahrenheit that the steam in the boiler was superheated	8.55
Temperature in degrees Fahrenheit of the steam in the steam pipe after the steam had passed through the reducing valve, by thermometer	335.54
Temperature in degrees Fahrenheit of the steam in the steam pipe after the steam had passed through the reducing valve, considered as saturated	329.93
Number of degrees Fahrenheit that the steam in the steam pipe was superheated	5.61
Temperature in degrees Fahrenheit of the gases of combustion in the chimney	Zinc barely melted.

Difference between the temperature of the steam in the boiler by thermometer, and the temperature of the steam in the steam pipe after the steam had passed through the reducing valve, by thermometer, in degrees Fahrenheit	17.88
Temperature in degrees Fahrenheit of the air in the fire room	99.16
Temperature in degrees Fahrenheit of the external atmosphere	51.08
Temperature in degrees Fahrenheit of the feed water entering the boiler	79.

RATE OF COMBUSTION.

Number of pounds of coal consumed per hour	437.5
Number of pounds of combustible consumed per hour	403.833333
Number of pounds of coal consumed per hour per square foot of grate surface	13.671875
Number of pounds of combustible consumed per hour per square foot of grate surface	12.619792
Fraction of a pound of coal consumed per hour per square foot of inner total heating surface	0.569663
Fraction of a pound of combustible consumed per hour per square foot of inner total heating surface	0.525826

VAPORIZATION.

Number of pounds of water that would have been vaporized in the boiler had it been supplied at the temperature of 100 degrees Fahrenheit and vaporized under the standard atmospheric pressure	90,452.023494
Number of pounds of water that would have been vaporized in the boiler had it been supplied at the temperature of 212 degrees Fahrenheit and vaporized under the standard atmospheric pressure	101,022.575645
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 100 degrees by one pound of coal	8.614478
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 100 degrees by one pound of combustible	9.332648
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 212 degrees by one pound of coal	9.621198
Number of pounds of water vaporized under the atmospheric pressure from the temperature of 212 degrees by one pound of combustible	10.423295
Number of pounds of water vaporized per hour under the atmospheric pressure from the temperature of 212 degrees by one square foot of inner total heating surface	5.48084

Making the same corrections in the same manner for the influence on the economic vaporization exerted by the temperature of the steam in the boiler during the experiment with the boiler of the *Shearwater*, that were made in the cases of the experiments with the boiler at the Delamater Iron Works given in tables Nos. 1 and 2, there result the following:

The temperature of the boiler steam considered as saturated, was 344.87 degrees Fahrenheit, the difference between which and 212 degrees is 132.87 degrees, consequently, the number of pounds of water vaporized from the temperature of 212° under the standard atmospheric pressure by one pound of the crude coal will be $(\frac{132.87}{1,800} + 1) \times 9.621198 = 10.331406$ pounds and by one pound of the combustible portion of that coal $(\frac{132.87}{1,800} +) \times 10.423295 = 11.192711$ pounds.

As the steam was superheated during the experiment with the *Shearwater's* boiler, it must have been still more superheated during the experiments in tables Nos. 1 and 2, so that any priming or foaming or entrainment of water in these cases was impossible.

The results of the three experiments, corrected for the influence of the different temperatures of the steam in the boiler on the economic vaporization of the coal are as follows:

1st. As regards the rate of combustion:

Number of pounds of semi-bituminous coal consumed per hour per square foot of grate surface in the experiment of Table No. 1	8.771053
Number of pounds of semi-bituminous coal consumed per hour per square foot of grate surface in the experiment of Table No. 2	9.201562
Number of pounds of semi-bituminous coal consumed per hour per square foot of grate surface in the experiment with the *Shearwater's* boiler	13.671875
Mean of the three	10.548163

Number of pounds of semi-bituminous coal, less the refuse, consumed per hour per square foot of grate surface in the experiment of Table No 1	7.875
Number of pounds of semi-bituminous coal, less the refuse, consumed per hour per square foot of grate surface in the experiment of Table No. 2	7.921875

Number of pounds of semi-bituminous coal, less the refuse, consumed per hour per square foot of grate surface in the experiment of the *Shearwater's* boiler	12.619792
Mean of the three	9.472222
Fraction of a pound of semi-bituminous coal consumed per hour per square foot of heating surface calculated for the inside dimensions of the tubes and connections in the experiment of Table No. 1	0.328126
Fraction of a pound of semi-bituminous coal consumed per hour per square foot of heating surface calculated for the inside dimensions of the tubes and connections in the experiment of Table No. 2	0.330079
Fraction of a pound of semi-bituminous coal consumed per hour per square foot of heating surface calculated for the inside dimensions of the tubes and connections in the experiment with the *Shearwater's* boiler	0.525826
Mean of the three	0.394677
Per centum of the coal in refuse, experiment of Table No. 1	10.216022
Per centum of the coal in refuse, experiment of Table No. 2	13.907285
Per centum of the coal in refuse, experiment of the *Shearwater's* boiler	7.695238
Mean of the three	10.606182

2d. As regards the economic vaporization. This vaporization in the different experiments is given for what it would have been had the evaporation taken place under the actual atmospheric pressure instead of under the experimental steam pressure, and had the feed water been supplied at the temperature of 212 degrees Fahrenheit.

Number of pounds of water vaporized by one pound of crude coal, Table No. 1	9.425299
Number of pounds of water vaporized by one pound of crude coal, Table No. 2	10.070563
Number of pounds of water vaporized by one pound of crude coal, *Shearwater* trial	10.331406
Mean of the three	9.942423
Number of pounds of water vaporized by one pound of combustible, Table No. 1	10.497752
Number of pounds of water vaporized by one pound of combustible, Table No. 2	11.697346

Number of pounds of water vaporized by one pound of combustible, *Shearwater* trial	11.192711
Mean of the three	11.129270

By "combustible" in the above is meant what remains of the crude coal after deduction of its refuse of ash, clinker, dust, etc.

3d. As regards the rate of vaporization:

Number of pounds of water vaporized per hour under the atmospheric pressure from the temperature of 212 degrees Fahrenheit by one square foot of inner total heating surface in the experiment of Table No. 1	3.446318
Number of pounds of water vaporized per hour under the atmospheric pressure from the temperature of 212 degrees Fahrenheit by one square foot of inner total heating surface in the experiment of Table No. 2	3.861047
Number of pounds of water vaporized per hour under the atmospheric pressure from the temperature of 212 degrees Fahrenheit by one square foot of inner total heating surface in the experiment of the *Shearwater's* boiler	5.885419
Mean of the three	4.397595

The above results are corrected for the difference due to the different experimental steam pressures.

The number of Fahrenheit units of heat transmitted per hour per square foot of the total inner heating surface, from the gases of combustion to the water at the temperature of 212 degrees Fahrenheit and under the atmospheric pressure, is ($4.397595 \times 965.7 =$) 4,246.76.

The mean temperature of the gases of combustion in the base of the chimney, was, during the three experiments, probably about 700 degrees Fahrenheit.

The steam superheating probably averaged about 12 degrees Fahrenheit in the boiler, and insured perfectly dry saturated steam delivered into the cylinder.

In a previous communication to this journal (Vol. II, No. 2,) the writer pointed out some differences between pipe boilers, such as the Belleville, and the regular marine cylindrical tubular boiler, and said that both could be made by proper proportions to give

equal economic vaporizations under equal conditions, but omitted to point out that one of these conditions required all the air entering in both cases to enter at the furnace only. Now, in the case of the cylindrical tubular boiler with its iron cylindrical furnaces *inside* the iron cylindrical shell, this condition is perfectly fulfilled; not so, however, in the case of the pipe boiler, unless its envelope of sheet iron is made air tight, so as to prevent air from entering directly into the space containing the heating surface or pipes. Practically, this sheet iron envelope or casing is never air tight, and the air pours into the space containing the heating surface at every seam and terminal edge of the casing, the thermal effect of which is that the cold inleaked air mixes with the hot gases of combustion, correspondingly reducing the temperature of the latter, and to that extent lessening their heating effect, besides decreasing the velocity of the chimney draft. The temperature of the gases of combustion is reduced in a larger proportion than is due to the mere lower temperature of the inleaked air, because the specific heat of that air is greater than the mean specific heat of an equal weight of the gases of combustion. Unless, therefore, especial care is taken to make the casing of the pipe boiler air tight, it will, under equal conditions otherwise, give a less economic vaporization than the marine tubular boiler, and it will likewise give a less potential vaporization, owing to its less velocity of draft. All experiments known to the writer show these facts, and the loss thereby caused may be, and, indeed, generally is considerable. The higher the temperature of the gases of combustion in the space containing the pipes, the greater will be the velocity with which the external air will rush in at the seams and edges of the casing of the pipe boiler.

As vaporization is effected in the boiler by the difference between the temperatures of the water and of the gases of combustion, whatever tends to diminish that difference—as mixing these hot gases with cold air, for example—tends to reduce the vaporizing efficiency of the unit of heating surface: the same metallic surface will, in equal time, vaporize less water the less the difference of temperature on its opposite sides, and this loss of economic

effect is additional to the loss by the heating of the inleaked air and by the heat transmuted into the power expelling the latter from the chimney into the atmosphere.

Further, the loss of heat by external radiation is much greater under equal conditions of rate of combustion, steam pressure, etc., in the case of the pipe boiler than in that of the marine tubular boiler. In the latter, the radiating metal of the shell is in contact on the inside with only the water and steam held by the boiler, and can, therefore, have only the radiation due to the difference between their temperature and the temperature of the external air; while, in the case of the pipe boiler, the radiating metal of the casing is in contact on the inside with the hot gases of combustion, and will, therefore, have the radiation due to the difference between their temperature and the temperature of the external air. The loss of heat by radiation will, therefore, be several times as much from the pipe boiler as from the marine tubular boiler. This loss can, indeed, in the case of both boilers be greatly reduced in *absolute* quantity by placing over each a covering of so-called non-heat-conducting material, the best of which, however, transmits heat in considerable quantity, but such a covering does not change the *relative* loss of heat in the two cases. And with the coverings and thicknesses practicable, this loss, as an absolute quantity, is sufficient to materially modify the relative economic vaporization of the two kinds of boiler in favor of the marine tubular.

When pipe boilers are set in brick masonry, the same objection still continues, and is, in fact, intensified. The bricks are very permeable to air; they will indeed freely pass water, besides which the contact between the bricks and the mortar is such that air enters in easily at the joints. There are also large air leakages wherever the iron work of the boiler lies against the brick masonry, due to the continual and different expansions and contractions of the metal and masonry. All of these causes combined allows so great a leakage of air directly into the space occupied by the pipes or heating surface, that the economic vaporization is often largely lessened thereby.

The preceding experiments with the Belleville boiler show an

economic vaporization, under the experimental conditions, fully 10 per centum less than is obtained from the cylindrical marine boiler with tubes returned above the furnaces, and having the same proportions of grate surface to heating surface, and being used under the same conditions. Other pipe boilers show the same inferiority in this particular, and it is a very serious one, so serious, indeed, as to probably preclude the adoption of this species of boiler for sea-going vessels, in which economy of coal is of the first importance, not only on account of the cost of the coal, but, still more, on account of its weight to be carried and the space required for its storage. When a boiler gives one-tenth less economic vaporization than another, one-ninth more of it is required to furnish the same weight of steam in equal times with equal rate of combustion of the same fuel, than if its economic vaporization were equal to that of the other boiler, involving, of course, one ninth more weight and bulk. That is to say, one-ninth more grate surface and heating surface is required with the pipe boiler than with the marine boiler having tubes returned above the furnace, to produce the same weight of steam per hour, other things being equal.

Further, for equal endurance and speed of vessel, one-ninth more coal must be carried in the bunkers, requiring proportionally greater bulk for the storage of this greater weight, so that whatever may be gained in the less weight and bulk of the pipe boiler, for equal grate surface and heating surface, over the marine boiler, will be much more than lost in the greater weights and bulks involved in the case of sea-going steamers, estimating coal as well as boiler, for equal effects. For yachts, launches, torpedo boats and similar small vessels, whose steaming is of rarely more than a few hours' duration, and in which the coal carried is only sufficient for that time, the advantage of less weight and bulk in the boiler alone may more than counterbalance the disadvantage of its lesser economic vaporization. Also, for sea-going steamers, when the only test, as in the case of naval vessels, is confined to a measured mile trial, or, at the most, to a trial of but a few hours, the less economic vaporization of the pipe boiler—the weight of coal carried in the bunkers

being a constant quantity with all kinds of boiler—enters as a factor in only a small degree, and, under these conditions, an ephemeral newspaper success may be obtained among non-professionals, but at the expense of forever afterwards heavily handicapping the vessel and diminishing its value for the long continued service it was constructed to perform.

In like manner, a steel or iron sea-going vessel unsheathed with wood coppered, does not show in a short trial with clean bottom any inferiority to the sheathed vessel, but the fatal omission will be felt during the whole after-life of the vessel in either a great reduction of speed by fouling, or a great increase of the power for equal speed. The conditions of maximum performance are very different for short trials and for long service, and to sacrifice the latter to the former, is charlatanry and not engineering.

Of course, numerous and lengthened experiments should be made on all kinds of pipe boiler to determine exactly their economic vaporization comparably with that of the regular marine boiler having its tubes returned above its furnaces. A final decision cannot be had from a few cases, but the mean from many may be depended on in this particular as well as in others. The last word should not be said until all the evidence, abundant and unimpeachable, is in, but the fact must be kept steadily in view that the use or non-use of pipe boilers, such as they are now known, will depend for sea-going vessels largely on this fact of economic vaporization, and, as up to this time the experiments are much too few to justify a verdict, only a provisional opinion should be permitted.

XXI.

NEW FORMS OF EVAPORATORS.

By Passed Assistant Engineer G. W. Baird, U. S. Navy.

The importance of the evaporator has been so much underestimated that it is still difficult to secure proper space for it on board ships. Placing the vertical evaporator between decks, without a hatchway above it, or even near it, has led to great inconvenience. Though the machine may fit into such a space, and the pipe connections may happen to be convenient to such a position, it should not be forgotten that on each square foot of its heating surface there is from six to ten times the quantity of water vaporized that was formerly obtained from an equal area of boiler-heating surface.

With these facts in view there is but one alternative; that is, to declare the importance of the machine and secure a convenient location for it, or so modify its form as to accommodate it to the spaces which are left for it. For a given weight, bulk, cost and minimum number of joints, the coil evaporator in a cylinder with its axis vertical, is the very best, but it requires considerable height. For ships already in use, or those nearly completed, having the most desirable space already allotted, it cannot reasonably be expected that the evaporator can be desirably located. I have therefore designed two sizes, cylindrical in form, but having their axes horizontal, and their surfaces composed of U-shaped tubes expanded into a cast-steel tube plate as shown in Fig. 1. It is convenient and quite light, and will be found easy to scale, as the tubes, with the steam chest may be hauled out together.

This machine conforms to the specifications of the Bureau of Steam Engineering in having no internally detachable joints, in having coils, and in having its longitudinal seam welded. Its heads, however, are of plate steel, and are made hemispherical to save weight and to avoid bracing. The tubes are of Tyler's

steel, extra heavy, and will stand the blows of a scaling hammer. The steam and drain pipes are so placed as not to interfere with the bonnets. Should a leak occur in a tube end or

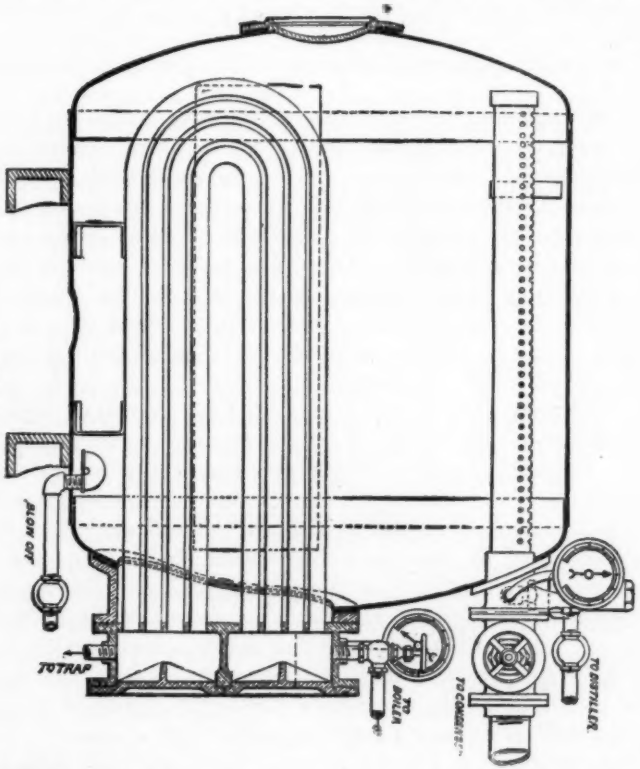


FIG. 1.

from a split tube it can be quickly located and plugged, for if the bonnets be removed while the evaporator is open to the condenser, the flame of a lamp placed near the tube end will quickly deflect towards and locate the leak. In scaling these tubes it is advisable to drive hard wood plugs into the tube ends to prevent their jarring, and possibly leaking. This cut represents a No. 3½ machine, which is 48 inches in diameter and 4

feet 8 inches long, and will make about 18 tons a day, blowing into a vacuum of 25 inches, or about 2,800 gallons a day working against the pressure in a distiller.

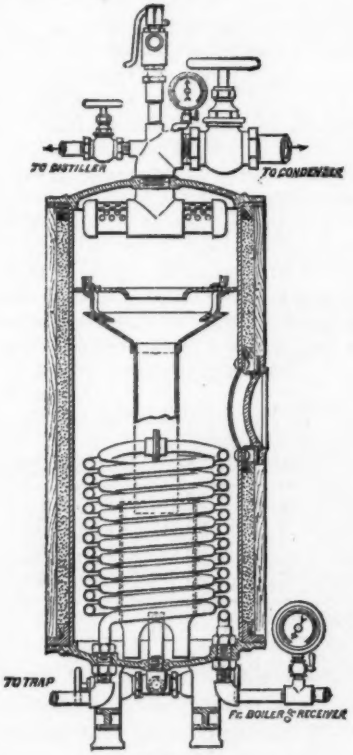


FIG. 2.

Figure 2 differs from the former vertical evaporators in having both ends of the coil (a double coil) pass through the lower head. By using stud bolts on the lower head, the shell may be detached without disturbing the lagging, and lifted off the coil, leaving the coil exposed for scaling. Three of these have been built; one for the *Vesuvius*, one for the *Pinta*, and one for the U. S. Revenue Cutter *Winona*. The last-named one was tested,

a few days ago, by Chief Engineer Wheeler, of the Revenue Marine Service, concerning which I submit the letter of the eminent engineer, Dr. Chas. E. Emery, and the table contained in that letter:

OFFICE OF CONSULTING ENGINEER, U. S. R. M.,
22 CORTLANDT STREET,
New York City, 13th Sept., 1890.

P. A. ENGINEER, G. W. BAIRD, U. S. N.,
Washington, D. C.

My Dear Sir: In response to your request for details of the test of your evaporator and distiller, I send herewith copies of the observations made by Chief Engineer Wm. C. Wheeler, U. S. R. M. I personally made some short tests, as checks, without taking the temperatures. The general result is that the evaporator, guaranteed to supply 500 gallons per day, showed a daily capacity of 1,200 gallons, and the distiller, guaranteed to supply 170 gallons per day, actually condensed at the rate of 345 gallons in that time. Very sincerely yours,

CHAS. E. EMERY,
Consulting Engineer, U. S. R. M.

TEST OF BAIRD'S EVAPORATOR.—28TH AUGUST, 1890.

Temperature of feed water.	Pounds of feed water per hour.	Weight of water discharged by the trap in lbs. per hour.	Temperature of water delivered by the trap.	Steam pressure in coils above atmosphere.	Steam pressure in evaporator above the atmosphere.	Pounds of water per hour from distiller.	Temperature of injection.	Temperature of discharge.	Temperature of distilled water.
78	408	434½	190	70	8	46	78	82	90

The evaporator was discharging against the atmospheric pressure through a ¾-inch opening, which accounts for the 8 pounds back pressure. This is the opening for the distiller; that to the main condenser is 2½ inches in diameter, and with this open, the vacuum in the evaporator should be nearly the same as in the condenser, when the output of the evaporator will be greatly increased, for the transmission of heat through the metal of the coils varies directly as the difference of temperature (and corresponding pressures) on the two sides of that metal.

By reference to the quantities obtained in this test it will be seen that the vaporization on the outside of the coils and the

condensation within very nearly balance. For example, the temperature in the coils (due to 85 pounds absolute pressure) was 316.1 degrees; the total heat from 0 degree is 1,209.0; the weight of water condensed in the coils was 434½ pounds, and the temperature of that water was 190 degrees. The thermal units are 434.5 (1,209.9 — 190 =) 443,146.5.

The pressure outside of the coils was 23 pounds absolute, the temperature 235.5 degrees, and the total heat was 1,185.2. The weight of water vaporized from 78 degrees was 408 pounds.

The thermal units in this case were 408 (1,185.2 — 78) = 451,737.6, or only about two per cent. more than the above, thus confirming the accuracy of the experiment.

It appears, however, that the weight of water condensed in the coils was greater than that vaporized on the outside by about 6 per cent. of the former. This will be different when exhausting into the condenser, for the reason that the total heat diminishes with the pressure.

The potential vaporization will rise considerably when blowing directly with the main condenser, owing to the greater difference of temperature on the two sides of the metal of the coils.

In using the evaporator on the main condenser it is advisable to employ, in the coils, the steam from the low pressure receiver, for the reason that it has done the greater part of the work, and its entire latent heat may still be utilized in vaporizing the water in the evaporator. The great advantage in using the evaporator directly on the condenser and supplying the coils with low pressure steam, is that the temperature of the sea water is kept below that at which there is any marked tendency to deposit sulphate of lime scale. Cousté's researches into the *rationale* of the formation of scale are well known, and the fact that he discovered, viz., that sulphate of lime is deposited by elevation of temperature beyond a certain point, has been put to practical use in the treatment of boilers using steam up to 80 pounds, where the amount of saline matter in the water was allowed to rise to four and four-and-a-half times that in ordinary sea water, with most beneficial results as contrasted with the former absurd

practice of maintaining the concentration at 1¾, which was always accompanied by enormous deposits of scale.

Prof. V. B. Lewes, in his recent admirable text book, "Service Chemistry," gives a table showing the effect of temperature on the amount of sulphate of lime that can be held in solution. It is so interesting that I reproduce it here.

Parts of sulphate of lime that can be dissolved in 100 parts of water at the following temperatures:

Degrees Fahrenheit.	Parts of sulphate of lime.
32	0.205
41	0.219
53.6	0.233
68	0.241
86	0.249
95	0.254
104	0.252
122	0.251
140	0.248
158	0.244
176	0.239
194	0.231
212	0.217
284 to 302	0.000

Now, if a vacuum of about 26 inches is maintained in the evaporator with a corresponding temperature of about 126° Fahrenheit, it is at once seen that there is scarcely any reduction in the solvent power of the water from the maximum at 95°. It would be hard to secure as good a vacuum as this, however, without special care, and we can see that even with only 22 inches and a temperature of about 153°, we have a loss of solvent power of only four per cent.

We have, however, not only the question of temperature to consider, but also that of the concentration of the water. Prof. Lewes shows that when boiled under atmospheric pressure (and 212° temperature), no sulphate of lime will be deposited until the density exceeds 1.09, or about three a half times the concen-

tration of ordinary sea water. Under the same conditions as to pressure and temperature, all the sulphate will be deposited when the density is about 1.225, or about eight and a third times the the concentration of ordinary sea water.

As the solvent action of water at 153° is greater than at 212°, we may reasonably expect that there will be no deposit until a concentration of from four to four and a half is reached, and very little up to about six.

We cannot hope to prevent entirely the deposit of sulphate of lime, but the method of working above outlined insures a minimum deposit at any density with economy in the output, or, if we are willing to sacrifice some of the economy, an almost entire prevention of deposit. As the evaporators are specially designed for facility in scaling, it amounts to saying that this method of working will reduce the frequency of scaling the coils enormously.

The system has not been working long enough yet to enable me to give definite figures, but the maintenance of the output at nearly the maximum for a much longer time than obtained with the older plan, shows that the deposition of the sulphate of lime has been reduced in a marked degree. There are several of my large evaporators now working successfully in this way on board the *City of Birmingham*, the *City of Washington*, and the *Seneca* and other vessels.

We are now building two large evaporators for Pacific mail steamers, and one for the Mallory line to be used in this way. They are to have brine pumps, but no feed pumps, as the feed water (which will come from the outboard delivery) will be drawn into the machine automatically. I have just completed a design for a distiller (condenser) to be used on shipboard, which has a combined air and circulating pump, which will enable us to extend this process to the manufacture of our potable water.

This, I feel, is the marking of an era in practical marine engineering. It will relieve us, I believe, of the great trouble of scaling boilers; and if we employ good filters for the feed water, and avoid the excessive use of cylinder oils, especially such as contain a quantity of acid, there is reason to believe our boilers

will last two or maybe three times as long as hitherto. This in itself is of great importance, for when we consider the lightness of scantling in the new boilers, and the amount of protective deck-beams, etc., to be removed in order to get a boiler out of a ship, we must appreciate the necessity of preservation.

With the new method of operating the evaporator we will be obliged to abandon our present hydrometers, and employ others which are graduated up to about $\frac{6}{32}$ and about 150 degrees. There will be, however, no difficulty in obtaining these.

It has been my purpose to deliver the water condensed in the coils at as low temperature as possible, in order that the pumps may work it without trouble. For this reason the steam is admitted at the top of the coils and discharged from the bottom, while the feed water is introduced at the bottom. By reference to Mr. Wheeler's test, it will be seen that steam entering the coil at a temperature of 316 degrees was condensed and discharged at 190 degrees. It is therefore apparent that, when the evaporator is discharging into a vacuum and with but one atmosphere of pressure in the coils, the drain water must be quite cool enough to be handled by any feed pump.

The question may present itself as to whether a donkey boiler may not at once displace the evaporator. But a moment's reflection will recall to us that, as the temperature of the furnace is so great, scale probably would be precipitated on the crown sheets, even though the *mean* temperature of the water in the boiler were no greater than in the evaporator. This can readily occur in those parts of the crowns where the circulation is sluggish. Another reason is that the boiler, from low water, would inevitably be burnt. The evaporator never can be.

The weight of a boiler, of equal potential, is greatly in excess of that of an evaporator, while a given area of heating surface in the latter is more than six times as efficient as the same amount of surface in the boiler.

Operating the plant in the new way forbids the use of the aerator, except to a very limited extent; but as the water is vaporized at a lower temperature it will not require so much air. It may not be out of place here to state that the artificial aerat-

ing of the water is not entirely for the purpose of making it "sharp" (like soda water), but to oxidize organic matter, which comes over with the steam, and the higher the temperature of that steam the more oxygen it requires. If, however, it is found that more air is necessary that can be used without too far vitiating the vacuum, we will still have the alternative of keeping sufficient storage in the tanks to enable the water to aerate in those reservoirs; for in the modern ships steam will be kept up continuously, and the daily consumption of water may be replenished with equal regularity. But we cannot hope to acquire a great gain without some loss. The elements of nature are constant in quantity, and the utmost man can do in the pursuit of physical truth is to shift the constituents, sacrificing one if he would produce another.

ASSISTANT ENGINEER W. M. MCFARLAND, U. S. NAVY.—In connection with the foregoing paper of Mr. Baird, I think the opportunity should not be lost of adding some valuable quantitative data in regard to the performance of evaporators under varying conditions which is furnished by an excellent article in the Transactions of the Institute of Engineers and Shipbuilders, in Scotland, for 1888–'89, by Mr. Charles Lang, entitled "On Evaporation." The entire article will repay careful study, but in what follows, the effort has been to give the more important points.

The liberty has been taken in the tabular results of Mr. Lang's experiments of changing the two columns relating to evaporation per square foot of heating surface. In the experiments the evaporation itself was not measured, but was obtained by calculation. The amount of steam condensed in the tubes was carefully measured, and the heat transmitted to the salt water for vaporizing calculated from the difference between the total heat of the steam and the total heat of the water of condensation.

Mr. Lang then calculates the weight of water vaporized by dividing the heat transmitted per square foot per hour by the *latent heat* of the steam in the shell outside the tubes. Although

he does not explain, it seems likely from the very limited duration of the experiments (the longest being only about seventeen minutes and shortest only one minute,) that the measurements were not made until the water surrounding the tubes had all reached the temperature due to the pressure in the shell. If this was actually the case, his figures were correct for those conditions, but they would not represent the work of an evaporator in actual practice where the water is fed in at a much lower temperature and must be heated up to the temperature due to the pressure before it can be vaporized. Moreover, his method of calculation makes the number of thermal units needed for vaporization less and less as the pressure increases, because the *latent* heat decreases as the pressure increases, while the total heat and the number of thermal units necessary for evaporation from a constant feed temperature would necessarily increase.

I have therefore recalculated those columns on the basis of a feed temperature of 70 degrees Fahr., as representing an average value in everyday service.

Mr. Lang says:

"Peclet made some observations with a long, open tube, and obtained very good results, but his apparatus was defective, as will be seen from the following considerations:

"1st. To have any tube surface uniformly efficient, it is necessary to keep up the pressure uniformly from end to end.

"2d. The air which is always associated with the steam must be continually removed as well as the condensed water formed in the tube.

"3d. The proportion of air and water present in the steam is a minimum at the inlet end of the tube, and increases to a maximum at the outlet end, and therefore the efficiency of the tube decreases in inverse proportion to the distance from the inlet end.

"The best form of tube to satisfy the first two conditions would be a long cone with the steam entering at the wide end, but this would not meet the requirements of the third condition."

He then describes a common form of evaporator, similar to a

EXPERIMENTS ON AN EVAPORATOR WITH VARYING PRESSURES IN TUBES AND SHELL.

Number of experiment.	Absolute pressure in pounds per square inch.			Temperatures Fahrenheit.			Weight of water condensed in tubes—pounds.	Time occupied—minutes and seconds.	Pounds of water condensed per hour.	Heating surface used, in square feet.	Heat units required to vaporize one pound of water in shell from 70° Fahrenheit.	Heat units given up by one pound of steam in tubes.	Heat units transmitted per hour per square foot of heating surface.	Pounds of water vaporized per hour per square foot of heating surface.	Heat units transmitted per square foot per hour for 1° Fahrenheit difference.	Pounds of water vaporized per square foot per hour for 1° Fahrenheit difference.
	Tubes.	Return.	Shell.	Tubes.	Return	Shell.										
A 1	29	28	23	248.4	246.4	235.5	56	9–37	349.4	38	1,115.2	942.7	8,667	7.80	671	.604
A 2	"	26	20	"	242.3	228	"	5–16	837.9	"	1,112.9	946.8	15,897	14.30	778	.701
A 3	"	24	17	"	238.8	219.6	"	3–45	896.	"	1,110.3	951.3	23,436	20.80	773	.698
B 1	35	34	29	259.3	257.6	248.4	112	10–45	312.5	"	1,119.1	934.9	7,663	6.85	884	.792
B 2	"	31	24	"	252.2	237.8	"	8–45	768.	"	1,115.9	940.3	19,004	17.03	884	.792
B 3	"	28	18.5	"	246.4	233.5	"	5–06	1,317.6	"	1,111.7	946.3	32,804	29.51	910	.824
C 1	50	48	40	281	274.4	267.3	"	16–42	402.4	"	1,124.0	930.7	9,749	8.67	911	.633
C 2	"	45	33	"	268.7	253.1	"	7–00	960.	"	1,121.5	934.6	23,338	20.83	930	.889
C 3	"	41	28	"	260.8	243.7	"	3–81	2,138.9	"	1,114.5	939.4	52,343	46.97	1,036	.980
D 1	80	78	65	312	310.2	298	56	10–30	316.7	21.95	1,134.3	898.3	12,703	11.18	1,078	.847
D 2	"	75	56	"	307.5	288.2	"	5–22	625.1	"	1,131.3	901.	25,659	22.68	1,078	.953
D 3	"	65	36	"	298	259.3	"	3–11	1,533.1	"	1,122.3	910.5	63,592	56.66	1,206	1.075
E 1	105	90	35	331.3	324.2	297.5	"	1–38	2,185.4	"	1,114.8	927.5	90,341	80.44	1,187	1.034
E 2	"	98	72	"	320.1	292.7	"	6–38	506.5	"	1,137.2	885.3	19,449	17.97	1,098	.855
E 3	"	90	50	"	306.3	281.	"	3–12	1,047.3	"	1,132.7	888.3	42,172	37.66	1,098	.970
F 1	140	130	90	"	320.2	297.4	"	2–20	1,440	"	1,130.7	894.2	58,262	51.96	1,166	1.033
F 2	"	74	40	"	306.6	267.3	"	1–48	1,853.8	"	1,124.9	907.8	76,668	68.76	1,198	1.085
F 3	"	68	33	"	294.8	255.9	"	1–33	2,167.7	"	1,121.5	913.6	91,016	80.99	1,194	1.074
G 1	135	124	75	330.1	323.6	295.5	112	5–18	1,266.9	"	1,137.5	876.6	50,593	44.49	1,187	1.044
G 2	"	111	56	"	313.3	292.7	"	3–14	1,813.1	"	1,134.7	884.9	73,254	64.51	1,272	1.123
G 3	"	95	45	"	304.1	274.4	"	2–13	2,466.	"	1,127.1	896.7	100,673	89.31	1,272	1.128
G 4	"	81	33	"	293.8	250.4	"	1–10	3,101.5	"	1,119.8	907.4	128,214	114.56	1,282	1.145
H 1	165	150	85	366	358.3	316.1	"	4–21	1,543.3	"	1,139.9	866.6	60,938	53.96	1,221	1.071
H 2	"	138	64	"	348.3	296.9	"	2–56	2,290.9	"	1,134.0	876.6	91,489	82.68	1,324	1.168
H 3	"	113	53	"	336	284.7	"	2–01	2,679.	"	1,130.3	888.9	106,900	95.98	1,334	1.180
H 4	"	92	35	"	321.7	259.3	56	1–03	3,174.8	"	1,122.5	903.2	130,037	116.40	1,324	1.191

surface condenser with the tubes taking steam at one end and discharging at the other into a common chamber with the drain valve at the bottom. He points out that this is sure to be inefficient because the air will collect in some of the tubes, and there is no way to get rid of it, the tube so filled being practically useless for evaporative purposes.

This is obviated by making all the condensed steam pass through the lower row of tubes which thus form a drain for all the others. As these tubes receive only water or partly condensed steam, the pressure in them will be lower than in any other row of tubes and, in consequence, the greater pressure in the others forces out all the air and sends it along with the water out through the bottom row of tubes. In addition to this feature, the back or outlet ends of all the tubes except the bottom row were contracted by inserting a plug with a small hole. This insures a lower pressure in the outlet chamber.

"By this means the pressure in the casing at the back end is always considerably lower than at the front end, and therefore a continuous current must take place through *all* the tubes. Owing to this constant current it is impossible that any air or water can lodge in any of the tubes to retard their heating power.

"A single example will serve to show the gain in efficiency by this improvement. An evaporator of the old type containing 28 tubes with a total heating surface of 95 square feet evaporated on an average 1,750 pounds of water per hour, or 18.4 pounds per square foot per hour, using steam at 40 pounds pressure inside the tubes and 5 pounds pressure in the shell. An evaporator with the improved arrangement with 12 tubes and 38 square feet of heating surface, using the same steam pressures, evaporated 2,100 pounds per hour, or over 50 pounds per square foot per hour.

"I have said that the average performance of the old evaporator was 1,750 pounds per hour, but sometimes the amount made was much greater and at others very much smaller. This was, of course, due to an accumulation of air in a number of the tubes as already explained."

A number of experiments were made at Holm Foundry, Cathcart, to determine the relation between the heating surface and quantity of water evaporated at various pressures of steam.

The evaporator used in these experiments was one of Weir's patent, embodying the improvement already explained. Its rated capacity was 10 tons of fresh water per 24 hours. The shell consisted of a steel cylindrical vessel, 3 feet diameter by 4 feet 3 inches long. The heating surface was composed of 12 solid-drawn copper tubes, $1\frac{1}{2}$ inches external diameter, No. 10 B.W.G. thick, giving a total heating surface of 38 square feet.

"Steam was led to the tube casing from our ordinary shop boiler, which is constructed for a working pressure of 200 pounds. A separator, with water gauge, was placed between the boiler and evaporator to insure that only dry steam entered the tubes.

"Salt water having a density of 19 ounces of salt to the gallon (this is about $\frac{4}{3\frac{1}{2}}$, as measured by our naval hydrometer,) was used in the evaporator. The steam evaporated from the salt water could be led either to a condenser or blown direct to the atmosphere. When the condenser was being used the condensed water was frequently tested to see that no priming took place.

"Pressure gauges were arranged to show the pressures in the direct tubes, in the return tube, and in the steam space of the shell. The course adopted was to make six distinct sets of experiments, each set consisting of from three to five different experiments. During each set the pressure in the tubes remained constant, while the pressure in the shell was varied in each experiment. The trials were conducted in conjunction with Mr. Woodthorpe, of the Consultative Department of the Board of Trade, and each one was carefully performed twice, the mean time being taken by a chronometer.

"The actual weight of boiler steam condensed in the tubes was taken by a weighing machine. The total heat given up by each pound of steam condensed in the tubes is plainly the difference between the total heat of the steam entering the tubes and the total heat of the water leaving the drain valve.

"On comparing our results with those previously obtained,

we find that the best results hitherto published have been those of Peclet.

"Using a copper tube 137.8 feet long, 1.36 inches outside diameter, made into a coil, with steam at 45 pounds pressure admitted freely into one end of the tube, his highest result was 948 units of heat transmitted per square foot of heating surface per hour for 1 degree Fahr. difference of temperature."

In another experiment, using two coils of copper pipe 52.5 feet long, 1.36 inches diameter, he obtained a still higher result, viz., 1,120 units.

"These results have usually been looked upon as inaccurate, as being in fact too high, but our table shows that not only have we reached the same figures, but that many of our results are a good way ahead of Peclet's, his highest result being 1120, while our highest is 1,334."

The table on page 505 contains the observed data of these experiments and the calculations therefrom.

Mr. Lang also contributes some very interesting remarks, as a result of experiments to determine the relation between the rate of evaporation and the occurrence of priming:

"Let us first define the term 'priming,' for it is used with two very different meanings. First, there is priming in the ordinary acceptation, denoting a condition where, on steam being supplied to a steam engine there are visible signs of water being also carried over. In the second place, priming may be taken to denote that state where, though there is no external evidence, the nitrate of silver test discloses the fact that faint traces of salt have been carried over. It is with this latter meaning that we shall use the term. The experiments were conducted with the same apparatus as those already described. The method of procedure was as follows.

"A number of pressures for the shell were fixed upon, and the maximum weight of steam that could be taken off at each of those pressures without priming was carefully measured.

"By gradually raising the pressure in the tubes, and keeping the pressure in the shell constant by adjusting the outlet valve, we ascertained the point where the evaporator just began to show

signs of priming with the silver nitrate test. The pressure in the tubes was then slightly reduced, and after seeing that no priming was taking place, the quantity of steam being evaporated per hour was measured.

"On examining our results, it appeared that the weight of steam evaporated per hour was almost exactly proportional to the density of steam in the shell. We may put this result into a slightly different form, which seems to be the law regulating priming under different conditions of steam pressure.

"In any boiler with the same quantity of water (same height in gauge glass) the *volume* of steam that can be obtained without priming (that is, in a perfectly dry state,) is the same at all pressures, but, as the density increases with the pressure, it follows, that the *weight* of steam obtainable is proportional to the density.

"Thus, if W denote weight of steam at density d, and W_1 the weight at density d_1, we have

$$\frac{W_1}{W} = \frac{d_1}{d}."$$

XXII.

NOTES ON ANALYSIS OF ENGINE TRIALS.

By Assistant Engineer W. H. Allderdice, U. S. Navy.

At the risk of seeming to underestimate the value of Chief Engineer Isherwood's analysis of the experiments on the ferry-boat *Bergen*, as given in the August number of the Journal, the writer has prepared the following notes, with reference to certain points in which the methods of computation employed in that analysis appear to him to be inaccurate.

The points considered are the following, and are taken up in the order named:

I. The method of computing the condensation of working steam during expansion.

II. The method of computing the total horse-power developed by the working steam.

III. The method of accounting for the distribution of the total horse-power.

IV. The method of computing the thrust of the screw.

I. The Condensation of Working Steam during Expansion.

The total condensation of steam in the cylinder of an engine using steam expansively may be considered as composed of two parts; the first and usually the greater part being that which is due to the cooling effect of the metallic walls of the cylinder, the second that which is due to the loss of heat by the expanding steam while doing work, after communication with the source of heat has been shut off. In a non-conducting cylinder, the second part only would exist; we should have, in such a cylinder, no condensation at all during that portion of the stroke from the beginning to the point of cut-off, all the work performed up to that point being at the expense of the source of heat, and by the transfer from the source of an amount of

heat equivalent to the work performed; but beyond the point of cut-off, all work performed would be at the expense of the heat energy in the expanding steam, and this, at all ordinary temperatures, would result in the condensation of a portion of that steam. Calculating the amount of this condensation in a non-conducting cylinder, we shall be able to determine, by comparison with an actual engine, what proportion of the total condensation in the latter is due to the cooling effect of the cylinder walls, and what to the work performed during expansion.

In the calculation of the amount of condensation due to the work performed during expansion, tabulated as "*Number of pounds of steam liquefied per hour * * * * to furnish the heat transmuted into the power developed by the expanding steam alone*," Chief Engineer Isherwood employs the following method of computation;

"*For every* 789¼ *foot-pounds of work done by the expanding steam on the diagram, after the closing of the cut-off valve, one Fahrenheit unit of heat disappears, so that the division of the number of foot-pounds of work done per hour by the expanding steam alone by* 789¼ *gives a quotient which, divided by the latent heat of steam of the mean pressure of the expanding steam, will give the number of pounds of steam liquefied per hour.*"

The statement of this method is given in Chief Engineer Isherwood's article on the "Test of Worthington Pumping Engines," in the February number of the JOURNAL; also, in somewhat different phraseology, in the published reports of experiments on the *Leila*, *Siesta* and *Trenton*. It is inferred from this, and from verification of the arithmetical work, that the same method is employed in the analysis of the *Bergen* experiments, no statement of the method used being given in the latter article.

The error of this method of computation lies in the assumption that the heat transformed into work during expansion is all given up by the steam that is condensed, and none of it by the steam that remains uncondensed. But, as a matter of fact, we know that the uncondensed steam remaining in the cylinder at the end of the expansion has given up a portion of its energy;

its temperature and pressure have been lowered from what they were at the beginning of the expansion; its total heat per pound has been reduced; consequently, any method of computation of the amount of steam condensed during expansion which fails to take into account the loss of energy of the uncondensed steam, must give a result that is too large.

A nearly correct method would be the following:

Let W = total work during expansion per pound of working steam per stroke, expressed in thermal units.

" H_1 = total heat of one pound of steam at beginning of expansion.

" H_2 = total heat of one pound of steam at end of expansion.

" L = mean latent heat of one pound of steam during expansion.

" x = fraction of each pound of working steam condensed during expansion.

then

$$L x + (H_1 - H_2)(1 - x) = W,$$

or

$$x = \frac{W - (H_1 - H_2)}{L - (H_1 - H_2)}.$$

Expressed in the same terms, Chief Engineer Isherwood's method would give, $x = \frac{W}{L}$, a result evidently larger than that given by this method.

A small error in this method lies in the fact that it fails to take account of the heat given up during the later stages of the expansion by the water condensed during the earlier stages. A further objection to its use is that it involves the mean latent heat of the expanding steam, something which would require a most elaborate calculation to obtain. An allowable approximation, and the one adopted by Chief Engineer Isherwood, is to use the latent heat corresponding to the mean pressure during expansion. Employing this approximation, and neglecting the small error referred to, this method would appear to be sufficiently accurate for all ordinary applications.

Solutions of the problem are given by both Rankine and Clausius. Rankine, in his "Manual of the Steam Engine and Other Prime Movers," treating of "Liquefaction of Steam Working Expansively," Art. 283, gives "for the proportion borne by the condensed steam to the whole mass of steam and water," the following expression:

$$\frac{v-u}{v} = 1 - \frac{\tau}{H'}\left(J \text{ hyp. } \log \frac{\tau_1}{\tau} + v_1 \frac{dp_1}{d\tau_1}\right).$$

The nomenclature employed is the following:

v_1 = volume of one pound of steam at beginning of expansion.

v = volume of one pound of steam at any time during expansion.

u = volume occupied by the mass of steam, or of steam and liquid water, under consideration, at any time during expansion.

τ_1 = absolute temperature of the steam at beginning of expansion.

τ = absolute temperature of the steam at any time during expansion.

J = mechanical equivalent of one heat unit.

H' = latent heat of evaporation of one pound of steam, at any time during expansion, expressed in foot-pounds.

$\frac{dp_1}{d\tau_1}$ = rate of variation of pressure with temperature, at beginning of expansion.

Rankine's empirical formula, deduced from Regnault's experiments, for the relation between the pressure of saturated steam, in pounds per square foot, and the corresponding temperature, is—

$$\log p = A - \frac{B}{\tau} - \frac{C}{\tau^3},$$

from which, for pressures per square inch,

$$\frac{dp}{d\tau} = 144\, p \left(\frac{B}{\tau^2} + \frac{2C}{\tau^3}\right) \text{hyp. log } 10.$$

For the constants, Rankine gives—

$$A = 8.2591.$$
$$\log B = 3.43642.$$
$$\log C = 5.59873.$$

The table on the opposite page, giving the proportion of steam condensed during expansion for various initial and final temperatures, has been calculated by means of Rankine's formula. Required data was obtained from D. K. Clark's tables of the properties of saturated steam. "Joule's equivalent," 772 foot-pounds, was employed as the mechanical equivalent of one Fahrenheit unit of heat.

The tabulated results will be found in every case to be considerably less than those obtained by Chief Engineer Isherwood's method.

Clausius, in "The Mechanical Theory of Heat," chapter VI., section 12, gives the formula—

$$m = \frac{T}{r}\left(\frac{m_1 r_1}{T_1} - M c \log \frac{T}{T_1}\right),$$

in which

M = total weight of substance at beginning of expansion.

m_1 = weight of that part of substance in the condition of vapor, at beginning of expansion.

m = weight of that part of substance in the condition of vapor, at any time during expansion.

$M - m$ = weight of that part of substance in the condition of liquid, at any time during expansion.

r_1 = latent heat of evaporation, in thermal units, of unit weight of the substance, at beginning of expansion.

r = latent heat of evaporation, in thermal units, of unit weight of the substance, at any time during expansion.

T_1 = absolute temperature of the substance at beginning of expansion.

T = absolute temperature of the substance at any time during expansion.

c = specific heat of the substance in the liquid state.

The assumption is that the substance is wholly in the condition of vapor at the beginning of the expansion, so that we may put $m_1 = M$. Also, since from Regnault's formula for the specific heat of water,

$$c = 1 + 0.00004\, t + 0.0000009\, t^2,$$

TABLE SHOWING PROPORTION OF STEAM LIQUEFIED DURING EXPANSION IN A NON-CONDUCTING CYLINDER.

Pressure at cut-off, in lbs. per sq. in., abs.	Temperature corresponding, Fahrenheit.	Pressures and corresponding temperatures at release.									
		100 lbs., 327°.9	90 lbs., 320°.2	80 lbs., 312°.0	70 lbs., 302°.9	60 lbs., 292°.7	50 lbs., 281°.0	40 lbs., 267°.3	30 lbs., 250°.4	20 lbs., 228°.0	10 lbs., 193°.3
150	358°.3	.02732	.03406	.04126	.04933	.05830	.06866	.08076	.09568	.11468	.14672
140	352°.9	.02254	.02936	.03664	.04480	.05390	.06432	.07654	.09164	.11082	.14316
130	347°.2	.02012	.02698	.03430	.04252	.05165	.06214	.07442	.08958	.10888	.14134
120	341°.1	.01345	.02037	.02782	.03615	.04540	.05607	.06853	.08388	.10346	.13632
110	334°.6	.00555	.01260	.02017	.02866	.03808	.04892	.06160	.07717	.09706	.13040
100	327°.9	.....	.00962	.01724	.02577	.03526	.04617	.05892	.07460	.09460	.12816
90	320°.2	.....	.....	.00830	.01700	.02667	.03777	.05077	.06676	.08712	.12120
80	312°.0	.....	.....	.....	.00970	.01952	.03078	.04397	.06022	.08088	.11546
70	302°.9	.....	.....	.....	.....	.01026	.02177	.03522	05176	.07280	.10798
60	292°.7	.....	.....	.....	.....	.....	.01297	.02667	.04352	.06496	.10072
50	281°.0	.....	.....	.....	.....	.....	.....	.01652	.03372	.05564	.09210
40	267°.3	.....	.....	.....	.....	.....	.....	.....	.01830	.04090	.07852
30	250°.4	.....	.....	.....	.....	.....	.....	.....	.....	.02370	.06258
20	228°.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	.03898

NOTE.—The table assumes the steam to be dry saturated steam at the beginning of expansion. To apply it to a triple-expansion engine, find first the total amount of liquefaction for the whole range of expansion of the engine, from the pressure at cut-off in the first cylinder to the pressure at release in the last; subtract from this the amount of liquefaction for the range of expansion in the first and intermediate cylinders; the difference will be the amount of liquefaction in the last cylinder. In the same manner the amount of liquefaction in the intermediate cylinder may be found by subtracting the amount of liquefaction in the first cylinder from the amount of liquefaction in the first and intermediate cylinders.

c is seen to vary very slightly with the temperature, it may be assumed as constant and equal to unity without appreciable inaccuracy.

To obtain "the proportion borne by the condensed steam to the whole mass of steam and water," the formula may be placed in the form

$$\frac{M-m}{M} = 1 - \frac{T}{r}\left(\log\frac{T_1}{T} + \frac{r_1}{T_1}\right).$$

Passing to the nomenclature of Rankine, this becomes

$$1 - \frac{\tau}{H'}\left(J\log\frac{\tau_1}{\tau} + \frac{H'_1}{\tau_1}\right).$$

But the volume of one pound of saturated steam (neglecting the volume of the liquid water) is shown by Rankine to be

$$v = \frac{H'}{\tau\frac{dp}{d\tau}}$$

For $\frac{H'_1}{\tau_1}$ may then be substituted its equivalent $v_1\frac{dp_1}{d\tau_1}$, giving for the fraction $\frac{M-m}{M}$ the value—

$$1 - \frac{\tau}{H'}\left(J\log\frac{\tau_1}{\tau} + v_1\frac{dp_1}{d\tau_1}\right),$$

which is identical with Rankine's expression.

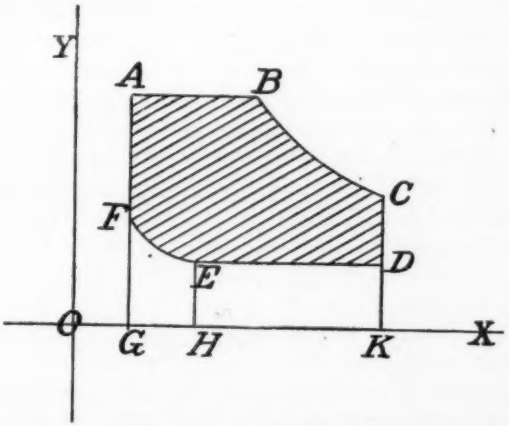
It may thus be seen that Rankine and Clausius, following quite different processes of investigation, arrive at the same result. It is pointed out by both authors that for some high initial temperature (calculated approximately by Rankine at about 1000°, Fahr.), the condensation during expansion would become zero, and for still higher initial temperatures, superheating would be produced by the expansion of working steam. It may be observed in comparison that Chief Engineer Isherwood's method would give positive values for the amount of the condensation, in all cases.

II. The Total Horse-Power of the Working Steam.

The method employed by Chief Engineer Isherwood for calcu-

lating the total horse-power of the working steam is stated as follows:

"*The total pressure on the piston of the small cylinder is the indicator pressure plus the back pressure against the piston above zero, exclusive of the cushioning, or for the length of the stroke of the piston between the end of the stroke and the point where the cushioning begins, and reduced in the ratio of that length to the whole length of the stroke. The total pressure so obtained is the proper pressure to use with the entire area of the piston of the small cylinder, and its speed, for calculating the total horses-power developed in the small cylinder due to the weight of feed-water pumped into the boilers, as this method eliminates the effect of the cushioning, the weight of steam cushioned or compressed remaining in the cylinder and not being discharged with the exhaust.*"



This is equivalent to saying that in an indicator diagram, *ABCDEF*, in which *BC* is the curve of expansion and *EF* the curve of compression, the axes *OX* and *OY*, representing respectively the lines of no pressure and no volume, the indicated horse-power per stroke being represented by the area *ABCDEF*, the total horse-power per stroke due to the weight of feed water pumped into the boilers, that is to say, the total horse-power per stroke developed by the working steam, will be represented by

the area *ABCKHEF*, which is the difference between the area *ABCKG*, representing the total horse-power per stroke developed by all the steam in the cylinder, and the area *FEHG*, representing the horse-power exerted per stroke in the cylinder in compressing the cushion steam.

The error of this method lies in the failure to eliminate entirely the action of the cushion steam, in order to study the action of the working steam.

To quote again from Rankine,—in "The Steam Engine and Other Prime Movers," Art. 262, on "Fluid Acting as a Cushion," he says:

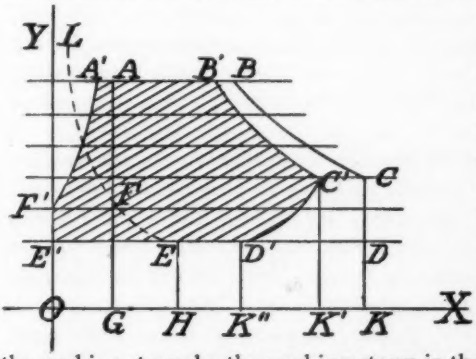
"To determine geometrically the efficiency of a heat engine, it is necessary to know its true indicator diagram; that is to say, the curve whose co-ordinates represent the successive volumes and pressures which the fluid *working the engine* assumes during a complete revolution. This true indicator diagram is not necessarily identical in figure with the diagram described by the engine on the indicator card, for the abscissæ representing volumes in the latter diagram, include not only the volumes assumed by that portion of the fluid which really performs the work * * * *, but also the volumes assumed by that portion of the fluid, if any, which acts merely as a *cushion* for transmitting pressure to the piston, undergoing, during each revolution, a series of changes of pressure and volume, and then the same series in an order exactly the reverse of the former order, so as to transform no heat permanently to mechanical energy."

Rankine's method of eliminating the action of the cushion steam, to obtain the true diagram of the working steam is employed by Professor Osborne Reynolds, in his article, "On the Triple-Expansion Engines and Engine Trials at the Whitworth Engineering Laboratory, Owens College, Manchester," published in the Proceedings of the Institution of Civil Engineers, Vol. XCIX., Paper No. 2407, and may be simply explained by a quotation from that article:

"To reduce this diagram, to show the relation of volume and of pressure of the steam discharged from the cylinder, an ideal compression-line is drawn through the point of the actual com-

pression-curve which corresponds to the closing of the exhaust. Horizontal lines are next drawn across the diagram, cutting the expansion-curve, the compression-line, and the ideal line, and each of these horizontal lines is set back until the point which was the ideal compression-curve reaches the line of zero volume. Then the position taken by the points from the expansion-line and the actual compression-line show the volume of steam in the cylinder over and above the volume of that which is shut in at exhaust. All this may be done arithmetically, or by plotting. The result is that, while the area of the diagram has not been altered, the actual expansion and compression-line for the steam passing through the engine is obtained."

Applying this method to the diagram previously considered, the curve *EFL* being the ideal compression curve drawn through the point *E*, at which the exhaust closes, we obtain the figure



A'B'C'D'E'F', as the true diagram of the steam passing through the cylinder, or working steam. It may be observed that the cycle of operations performed by and on the cushion steam, inclusive of the compression of the cushion steam by the working steam in the steam pipe before entering the cylinder, has been completely eliminated, and the resulting diagram represents solely the series of volumes and pressures of the working steam. The indicated horse-power per stroke is represented by the area *A'B'C'D'E'F'*, which is obviously equal to *ABCDEF*, the original indicator diagram. The total horse-power per stroke is represented by the area *A'B'C'K'OF'*, which differs from the area *ABCKHEF*, representing the total horse-power per stroke according to Chief Engineer Isherwood's method, by the amount of the small area

C'D'K''K'. This small area represents the total horse-power per stroke developed by the cushion steam in expanding from the final forward pressure to the back pressure, during which expansion it performs work in pushing out the exhausting steam against the existing pressure of that steam. By Chief Engineer Isherwood's method, this work performed *by* the cushion steam is not taken into account as forming part of the total work, yet the equivalent amount of work done *on* the cushion steam, in compressing it, is subtracted from the total work performed by both working and cushion steam, consequently the result obtained by that method must be too small in all cases in which the back pressure is lower than the final forward pressure.

III. The Distribution of the Total Horse-Power.

The method employed by Chief Engineer Isherwood for calculating the total horse-power developed in the intermediate cylinder, and that developed in the large cylinder, is stated as follows:

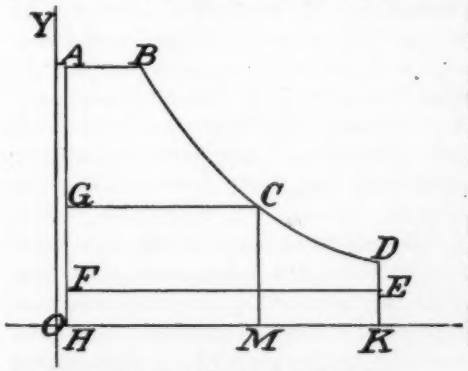
"*The indicated and net horses-power developed in the intermediate cylinder are calculated for the respective indicator and net pressures for the entire area of the piston; but the total horses-power therein developed are calculated for only the annular surface which remains of the piston after subtracting from the entire area of the piston the area of the piston of the small cylinder. This subtraction is required because the back pressure against the piston of the small cylinder is included in the back pressure against the piston of the intermediate cylinder, having been taken down to the zero of pressure. The total pressure on the annular space just described of the piston of the intermediate cylinder is the indicated pressure plus the back pressure, exclusive of the cushioned steam reduced in the ratio of the fraction of the stroke of the piston before the cushioning commenced to the entire stroke.* * * * * *The powers developed in the large cylinder have been calculated in a similar manner as for the intermediate cylinder.*"

That this "annular ring" method is correct when the cylinders

are in constant communication from beginning to end of stroke, one cylinder exhausting directly into the other, admits of no question; but it may be shown by two simple cases that its application is not general.

In the first place, suppose a compound engine, without compression in either cylinder, with the final forward pressure in the first cylinder the same as the back pressure in that cylinder, and with clearance spaces so small that their influence may be neglected; and suppose, for simplicity only, that there is an intermediate receiver between the cylinders so large that the pressure therein is practically constant. Steam being cut off in the large cylinder at a volume equal to the total volume of the small cylinder, there will be no "drop"; the friction in the steam passages may be neglected.

The combined card of such an engine would be represented by *ABDEF*, the total horse-power per stroke for the whole engine by the area *HABDK*, and the total horse-power per stroke for the small cylinder by the area *HABCM* (whether calculated by Chief Engineer Isherwood's method or by the method described in the previous section); consequently the total horse-power per stroke for the large cylinder would be represented by the area *MCDK*.



HK represents the volume of the large cylinder, and *HM* that of the small cylinder; therefore *MK* represents the volume obtained by multiplying the length of the stroke by the area of the annular surface which remains of the piston of the large cylinder after subtracting the area of the piston of the small cylinder. The mean ordinate of the area *HGCDK* represents "the indi-

cated pressure plus the back pressure" for the large cylinder. As there is no cushioning, the exclusion of the cushioned steam and reduction "in the ratio of the fraction of the stroke of the piston before the cushioning commenced to the entire stroke" is eliminated.

Now, it is at once apparent that the area *MCDK*, already shown to represent the total horse-power per stroke for the large cylinder, can *not* be obtained by multiplying *MK* by the mean ordinate of the area *HGCDK*; yet, if Chief Engineer Isherwood's method were correct, it would of necessity be equal to the product of these two quantities.

Again, we may conceive of a compound engine with two cylinders of the same size, without cushion in the second cylinder, and with cushion at, say, three-fourths of the stroke in the first cylinder. In such an engine the volume of steam discharged by the first cylinder would be three-fourths of the cylinder volume, and the same steam when discharged by the second cylinder would have been expanded one and one-third times—that is, to the full volume of the cylinder; consequently the total horse-power developed by the steam during this expansion should be credited to the second cylinder. Yet by Chief Engineer Isherwood's method we should obtain no total horse-power from this cylinder, since no annular surface would remain after subtracting the area of the first piston from that of the second, the two pistons being of the same size.

It is true that both of these cases are extreme cases, but it may be observed that the assumptions which make them extreme cases are introduced for the purpose only of simplifying the problem in question, and do not alter the underlying principles. They approach near enough to the practical to show that a method of calculation which is clearly inapplicable to them cannot be accurate when applied to the analysis of the trials of actual engines.

The best method of accounting for the distribution of the total horse-power in the three cylinders of a triple-expansion engine would appear to be to correct the three indicator diagrams in the manner previously pointed out, eliminating the

action of the cushion steam, and obtaining true diagrams of the working steam; combining these, we should obtain a diagram showing the complete action of the working steam. On such a diagram the total horse-power developed per stroke by the whole engine would be represented by the over-all area, measured down to the line of no pressure; removing from this area the area representing the total horse-power developed in the small cylinder, we should have remaining an area representing the total horse-power developed in the intermediate and large cylinders; removing in turn from this area the area representing the total horse-power developed in the intermediate cylinder, we should have remaining an area representing the total horse-power developed in the large cylinder.

IV. The Thrust of the Screw.

In accounting for the "distribution of the power" in order to obtain from the "horses-power expended in the propulsion of the vessel," the amount of the thrust of the screw, Chief Engineer Isherwood uses the *apparent* slip in the calculation of the "horses-power expended in the slip of the screw." Such a method must give at times anomalous results. An instance may be found in the published report of the *Trenton* experiments, made in 1883, where, in one case, the ship being under sail, no slip was observed; the "power expended in the slip of the screw" is tabulated as 0.0000, and the "power expended in the propulsion of the vessel" as 603.5781. The two results are manifestly inconsistent.

We cannot measure the *real* slip, and the value of calculations of the thrust based on *apparent* slip would seem at the best to be doubtful.

XXIII.

THE CONTRACT TRIAL OF THE U. S. S. *PHILADELPHIA.*

By Assistant Engineer W. H. Chambers, U. S. Navy.

The *Philadelphia* is a twin-screw steel cruiser with protective deck, built by Wm. Cramp & Sons, Philadelphia. There are three masts fitted with fore-and-aft sail.

The main battery consists of twelve VI-inch B. L. R., mounted four on each side of the main deck in projecting sponsons, two on the forecastle and two on the poop. The secondary battery consists of four 6-pounders, four 3-pounders, three 37-mm., two 1-pounders and four Gatlings.

The contract required an average speed of 19 knots for four consecutive hours, subject to a penalty of $50,000 for each quarter of a knot below and a premium of $50,000 for each quarter of a knot above the contract speed.

The following are some of the principal dimensions and data of the hull:

Length, extreme	335 feet.
Length between perpendiculars	315 feet.
Beam extreme	48 feet 6⅛ inches.
Beam at L. W. L	48 feet 6¼ inches.
Ratio of length to breadth	6 49 to 1.
Depth in hold from main deck beams to top of floors	31 feet 1.8 inches.
Displacement, mean for trial	4,325 tons.
Draught, mean for trial { Forward, 16 feet 10 inches / Aft, 21 feet 7 inches }	19 feet 2½ inches.
Displacement per inch at L. W. L	25.89 tons.
Area of L. W. L. plane	10,875 square feet.
Area of immersed midship section, on trial	815 square feet.
Coefficient of fineness on midship section	.597
Coefficient of fineness on L. W. L	.712
Coefficient of fineness on extreme dimensions	.515
Transverse metacenter above C. of G	2.1 feet.
Moment to alter trim one inch	420 foot-tons.
Normal coal supply	400 tons.
Maximum capacity of bunkers	850 tons.

MACHINERY.

The machinery was designed by the builders. There are two horizontal direct-acting triple-expansion engines in separate water-tight compartments, the forward engine driving the port propeller. The H.P. cylinder of the starboard engine is forward and the propeller right handed. The H.P. cylinder of the port engine is aft and the propeller left handed. The cylinder linings are jacketed. The main steam valves are of the piston type and single ported. There are two for each L.P. cylinder, two for each I.P. cylinder and one for each H.P. cylinder, all operated by the Marshall valve gear with rock shafts. The movement of each valve is regulated by a reversing arm and a radius link, and the cut-off varied by changing the position of a link block in the end of the reversing arm. The reversing engine is a single vertical cylinder with a floating lever for controlling its valve. It has a horizontal hand lever, similar to that of the main steam throttle and placed conveniently near it. There is no arrangement for reversing by hand. The main steam throttle is a double disc valve, opening with the pressure in the steam pipe. The pistons are cast-steel in dished shape with interchangeable rods of forged steel. The connecting rods are forged steel with composition boxes for the crank pin and crosshead. The crank pin brasses are lined with anti-friction metal. The crossheads are forged steel, fitted with cast-steel slippers, lined with anti-friction metal. The guides are cast-steel, hollow for circulation of water, and act as ties between the cylinders and the main crank shaft bearings. The castings for the crank shaft bearings are steel, hollow and in five pieces, one for each bearing. They are bolted together and secured to the cylinders by steel tie rods and the engine bed frames. The crank shafts are forged steel, built up with cranks at equal angles and with the couplings forged on. The main bearings have composition boxes and steel caps, both lined with anti-friction metal, and the boxes are hollow for the circulation of water. The pins have telescopic and centrifugal oilers. The propeller, line and thrust shafts are forged steel with axial holes. The forward lengths

of propeller shafts are cased with brass in the stern pipes, and the after lengths at the hanger bearings only.

The main condensers are cylindrical, of composition, cast in sections. Each main condenser has two vertical single-acting air-pumps, operated by an inclined compound engine. There is an overhead shaft with two cranks, the connecting rods of the pumps and steam cylinders engaging the same pins. A crank at one end of the shaft operates a double cylinder bilge pump. There is a centrifugal circulating pump for each main condenser, operated by a horizontal single cylinder engine.

The auxiliary condenser is composition, vertical and cylindrical. It is placed on the port side of the forward engine room, and has an independent air and circulating pump. A vertical inverted steam cylinder operates a vertical air pump and a centrifugal circulating pump.

Each main engine is provided with a separator and governor.

The following are some of the principal dimensions and ratios of the machinery:

Number of cylinders (each engine)	3.
Diameter of H. P. cylinder	38 inches.
Diameter of I. P. cylinder	58 inches.
Diameter of L. P. cylinder	86 inches.
Stroke, common	40 inches.
Diameter of piston rod	7½ inches.
Diameter of H. P. valves (one)	21 inches.
Diameter of I. P. valves (two)	21 inches.
Diameter of L. P. valves (two)	32 inches.
Area of H. P. steam ports, maximum opening	Outboard, 98.9 square inches. Inboard, 137.3 square inches.
Area of I. P. steam ports, maximum opening	Outboard, 197.9 square inches. Inboard, 274.6 square inches.
Area of L. P. steam ports, maximum opening	Outboard, 326.8 square inches. Inboard, 406.4 square inches.
Area of H. P. exhaust ports, maximum opening	177.2 square inches.
Area of I. P. exhaust ports, maximum opening	354.4 square inches.
Area of L. P. exhaust ports, maximum opening	570.6 square inches.
Area of main steam pipe (diameter 15½ inches)	188.7 square inches.
Area of main exhaust pipe (diameter 30 inches)	706.9 square inches.
Volume swept by H. P. piston per stroke, net	25.74 cubic feet.
Volume swept by I. P. piston per stroke, net	60.65 cubic feet.
Volume swept by L. P. piston per stroke, net	133.97 cubic feet.

Volumetric clearance in H. P. cylinder, per cent., about ..... 32.
Volumetric clearance in I. P. cylinder, per cent., about ..... 24.
Volumetric clearance in L. P. cylinder, per cent., about ..... 20.
Ratios of net areas of pistons ..... { H. P. to I. P ..... 1 to 2.36. / I. P. to L. P ..... 1 to 2.21. / H. P. to L. P ..... 1 to 5.20.
Lengths of connecting rods between centers ..... 86 inches.
Diameters of connecting rods ..... 7 inches and 6 inches.
Diameters of crank pins ..... 15 inches.
Diameters of crank shafts ..... 14½ inches.
Diameters of holes in pins and shafts ..... { After ..... 6½ inches. / Middle ..... 7 inches. / Forward ..... 7½ inches.
Lengths of pins ..... 17 inches.
Lengths of main journals, total ..... 121¾ inches.
Widths of crank webs ..... { Shaft end ..... 29 inches. / Middle ..... 25 inches. / Pin end ..... 28 inches.
Thickness of crank webs ..... { H. P ..... 9½ inches. / I. P. and L. P ..... 10½ inches.
Diameter of line shafts ..... 13¾ inches.
Diameter of thrust ..... 13¾ inches.
Diameter of holes in above ..... 6 inches.
Diameter of propeller shafts ..... { Forward, 14½ inches. / After ..... 15 inches.
Diameter of holes in above ..... { Forward, 8 inches. / After ..... 9 inches.
Thickness of composition casing, at bearings ..... 1⅛ and 1 $\frac{5}{32}$ inches.
Thickness of composition casing, remainder ..... ⅜ inches.
Diameter of crank shaft couplings ..... 2 feet 4½ inches.
Thickness of crank shaft couplings ..... 3¾ inches.
Outside diameter of thrust collars ..... 18 inches.
Inside diameter of thrust collars ..... 13¾ inches.
Number of thrust collars ..... 12.
Width of thrust collars ..... 1⅜ inches.
Space between thrust collars ..... 1¼ inches.
Thrust surface ..... 1,206 square inches.
Length of starboard shafting ..... 96 feet 3½ inches.
Length of port shafting ..... 123 feet 8½ inches.
Condenser:
Total length ..... 14 feet 1½ inches.
Diameter of shell, inside ..... 5 feet 8 inches.
Thickness of shell ..... ½ inch.
Length of tubes exposed ..... 12 feet.
Diameter of tubes, outside ..... ⅝ inches.
Number of tubes, one condenser ..... 3,440.

Cooling surface, one condenser..... 6,755 square feet.
Cooling surface, auxiliary condenser..... 68.19 feet.
Cooling surface used on trial..... 13,578.19 feet.
Air pumps:
Steam cylinders (2) diameters..... { H. P., 11 inches. L. P., 18 inches.
Pump cylinders (2) single-acting diameters..... 24 inches.
Stroke..... 18 inches.
Ratio of volume swept by L. P. piston per stroke to pir pump buckets per stroke..... 13.88 to 1.
Circulating pump:
Steam cylinders (1)..... diameter 11 inches, stroke 6 inches.
Diameter of runner..... 2 feet 8 inches.
Diameter of suction and discharge..... 15 inches.
Main bilge pumps:
Steam cylinders, same engine as air pump.
Pump cylinders (2), diameter plunger..... 4 inches.
Stroke..... 18 inches.
Propellers:
Type, modified Griffiths, with adjustable blades; material, manganese bronze.
Number of blades..... 3.
Diameter..... 14 feet 6 inches.
Pitch, limits of adjustment..... 18 feet 6 inches to 21 feet 6 inches.
Pitch, as set on trial..... { Starboard, mean, 20 feet 4.68 inches. Port..... mean, 20 feet 4.72 inches.
Ratio of diameter to pitch..... { Starboard, 1 to 1.4062 Port..... 1 to 1.4065
Disc area (blades and hub)..... 165.13 square feet.
Projected area..... 43.86 square feet.
Helicoidal area..... 57.1684
Greatest width of blade at radius of 3 feet 11 inches..... 53¼ inches.
Diameter of hub..... 3 feet 11½ inches.

BOILERS.

There are two main boilers in a forward water-tight compartment, and two main boilers and one auxiliary boiler in an after water-tight compartment, with athwartship coal bunkers forward of, abaft and between the compartments. The auxiliary boiler is placed in a recess in the after athwartship bunker, and is a cylindrical, single-ended, horizontal, return-fire tube boiler with two furnaces.

The main boilers are double ended, horizontal, return-fire tube boilers, with four furnaces at each end and a common back connection for each pair of opposite furnaces. The uptakes of each

pair of boilers unite above the protective deck and lead to a cylindrical standing smoke pipe, 60 feet in height above the upper grates and 6 feet 9 inches in diameter. The uptake of the auxiliary boiler leads into the after smoke pipe.

There is an air-tight bulkhead across each end of the main boilers, with a connecting passage on the port side of each pair. The passages between the water-tight compartments are amidships. There are four bunker doors in each fire room, two into the side bunkers and two into the athwartship bunkers.

There are eight blowers, two in each fire room, drawing air from ventilators and discharging directly into the fire rooms without air ducts. Each blower is driven by a brotherhood engine.

In each water-tight compartment there are two vertical duplex feed pumps—one main and one auxiliary. There is also an auxiliary feed pump for the auxiliary boiler.

The distilling apparatus is on the berth deck, and is designed to furnish 3,000 gallons of water in 24 hours.

The following are some of the principal dimensions and ratios of the boilers:

FOR ONE BOILER.

Length	19 feet 10⅞ inches.
Diameter	14 feet.
Number of furnaces in each (corrugated)	4.
Diameter of furnaces, greatest external	3 feet 4 inches.
Diameter of furnaces, least internal	3 feet.
Length of grate	6 feet 6 inches.
Outside diameter of tubes	2¾ inches.
Length of tubes between sheets	7 feet 7⅞ inches.
Number of ordinary tubes	796.
Number of stay tubes	168.
Thickness of ordinary tubes	No. 12 B. W. G.
Thickness of stay tubes	No. 8 B. W. G.
Depth of combustion chamber	4 feet 6 inches.
Height of combustion chamber above back end of upper grates	5 feet 8½ inches.
Height of combustion chamber above back end of lower grate	7 feet 0½ inch.
Capacity of furnaces and combustion chambers above grates and bridge wall	523 cubic feet.
Thickness of shell plates	1⅛ inches.
Thickness of tube sheets	$\frac{5}{16}$ inch.

Thickness of front head above tube sheet..... $\frac{7}{8}$ inch.
Thickness of combustion chamber..... $\frac{9}{16}$ inch.
Thickness of furnace..... $\frac{1}{2}$ inch.
Diameter of rivets..... $1\frac{1}{4}$ inches.
Number of braces above tubes..... 18.
Diameter of braces above tubes..... $2\frac{1}{2}$ inches.
Load on safety valve..... 160 pounds.
Test pressure..... 240 pounds.
Tube heating surface..... 4,400.4 square feet.
Plate heating surface..... 714 square feet.
Total heating surface..... 5,114.4 square feet.
Grate surface..... 156 square feet.
Area through tubes..... 21.3 square feet.
Area of water surface ($6\frac{3}{4}$ inches above tubes)..... 237 square feet.
Steam room ($6\frac{3}{4}$ inches above tubes)..... 587 cubic feet.
Weight of one boiler complete, without exterior or furnace fittings or water, 52.5 tons.
Weight of water in one boiler..... 35.4 tons.
Weight of water and boiler..... 87.9 tons.
Area of each smoke pipe..... 35.8 square feet.
Height of smoke pipe above upper grates..... 60 feet.

TOTAL FOR FOUR MAIN BOILERS.

Grate surface..... 624 square feet.
Heating surface tubes..... 17,601.64 square feet.
Heating surface plates..... 2,856.04 square feet.
Heating surface total..... 20,457.68 square feet.
Area through tubes..... 85.3 square feet.
Capacity of furnaces and combustion chamber above grates and bridge walls..... 2,092 cubic feet.
Steam room..... 2,348 cubic feet.
Area of water surface..... 948 square feet.
Weight in boiler compartments, with water..... 502.2 tons.
Blower engines, 3 cylinders, 5 inches diameter by 5 inches stroke.
Ratios:

Tube H. S. to G. S..... 28.20
Plate H. S. to G. S..... 4.58
Total H. S. to G. S..... 32.78
Area through tubes to G. S..... 1 to 7.3
Area of smoke pipes to G. S..... 1 to 8.72
Water surface to G. S..... 1.45
Steam room per square foot of G. S..... 3.76
Capacity of combustion chambers and furnaces per square foot of G. S..... 3.35
Weight in boiler compartment per 100 feet of H. S..... 2.39 tons.

AUXILIARY BOILER (NOT USED ON TRIAL).

Length..... 8 feet 3 inches.
Diameter (inside)..... 8 feet 6 inches.

Number of furnaces in each	2.
Diameter of furnaces, smallest internal	2 feet 8 inches.
Length of grate	3 feet 10 inches.
Grate surface	21.4 square feet.
Heating surface	531.1 square feet.
Ratio of grate surface to heating surface	1 to 24.82
Weight complete without exterior or furnace fittings or water	8.86 tons.
Weight of water	5.51 tons.
Weight, total	84.37 tons.

WEIGHTS.

Propelling machinery	698.33 tons.
Water in above machinery (6¾ inches above tubes)	153.29 tons.
Tools, instruments and duplicate pieces	16.22 tons.
Miscellaneous machinery	7.03 tons.
Total weight at time of completion of machinery	874.87 tons.

The trial took place June 25th, over a course previously laid out to the southward of Long and Block Islands. The sea was smooth and the wind from the W.S.W. with a force from 2 to 3. At about 8 A. M. the vessel got under way from off Fire Island, where she had anchored for the night, having come from Cramp's shipyard during the previous day. When about 10 miles from the course the speed was increased to 18 knots and the trial commenced at about 12.30 and ended at about 4.48 P. M.

COURSE DATA.

	Chronometer time. h.	m.	s.
Outer run, 1st transit	5	48	27
Outer run, 2d transit	7	54	49½
First elapsed time	2	06	22½
Inner run, 1st transit	8	08	17
Inner run, 2d transit	10	08	48
Second elapsed time	2	00	31
Total elapsed time	4	06	53½
Length of intended course (out and in)	80.292 miles.		
Outer run, as made	40.161 miles.		
Inner run, as made	40.271 miles.		
Length of course, as made	80.432 miles.		
Correction for tidal current	.538 miles.		
Length of corrected course	80.970 miles.		
Average speed per hour	80 970 ÷ 4.11479 = 19.678 miles.		

No patent log was used during the trial.

The correction for tidal current was made from observations by

vessels stationed at points on the course. These vessels were not all in place, but the observations were regarded as sufficiently accurate for making the correction.

The boilers steamed freely, furnishing throughout the trial all the steam required by the engines and auxiliaries without the use of the auxiliary boiler.

The engines ran smoothly and with the exception of the H. P. crank pin of the starboard engine and the I. P. crank pin of the port engine, on which it was necessary during part of the trial to run water, the journals showed no tendency to heat. As a precautionary measure small streams of water were run on the L. P. crank pin of the starboard engine and on all the line shaft journals. During the third hour the telescopic oiling arrangement of the I. P. pin of the port engine was broken and the indicator gear of that cylinder was disabled. The movement of the air pumps irregular.

There was no attempt made while off the course between the outer and inner runs to adjust the throttle or cut-offs, or to work the fires with a view to obtain better results on the inner run. It is estimated that the vessel ran a distance of about 100 miles while under the forced draft for the trial.

The following day the machinery was run for two hours with two boilers under forced draft to estimate the coal consumption, and the manœuvring qualities of the vessel were also tested.

After the return to Cramp's ship-yard the machinery was examined and found to be in good condition with the following exceptions: a very slight scoring of the pins and brasses of H. P. and L. P. crosshead journals of port engine and I. P. crosshead journal of starboard engine, slight leaks in the back connections of the boilers. These defects were regarded by the board of inspecting engineer officers as unimportant.

The following is a synopsis of the manœuvring tests, taken from the report of the Board:

EVOLUTION.

	Time. h.	m.	s.
Using wheel on top of pilot house (steam):			
Going ahead, full speed signal was made to back	5	17	40
Ship commenced going astern		18	40

Signal "Go ahead,"	19	
Going ahead	19	20
Order "Hard astarboard,"	21	50
Started to reverse port engine	22	12
Half way round (16 points,)	24	13
All the way round	27	31
Hard aport (reversing starboard engine, going ahead with port,)	27	44
Half way round (put helm amidships,)	32	07
All the way round	34	52
Full speed ahead	35	12
Wheel in conning tower (steam). Helm being hard astarboard, shifted helm to hard aport. Ship went off 6, and started to go to astarboard in 16 seconds.		
Order "Hard astarboard,"	38	13
Hard over	39	29
All the way around, order "Hard aport,"	43	52
Hard over	44	07
Order "Steady,"	44	32
Helm amidships	44	39
Wheel on quarter deck (steam):		
Order "Hard astarboard,"	51	25
Hard over	51	40
Order "Hard aport,"	51	47
Hard over	52	19
Order "Helm amidships,"	52	20
Helm amidships	52	38
Order "Hard astarboard,"	6 02	10
1½ points	3	20
Order "Hard aport,"	4	25
2½ points	5	14
Order "Helm amidships,"	6	20
Amidships	6	45

Ship going ahead full speed, took 9 minutes to change from steam to hand (quarter deck). Friction attachment secured by bolts only intended for emergency. With friction connection only, and with men accustomed to gear, it will probably take from ½ to 1 minute. Took about 1 minute to put it over 30°, wheel below aft (hand).

Synopsis of Official Speed Trial.

AVERAGES.	Starboard.	Port.
Steam at boilers (per gauge)	159.8	
Steam at engines (per gauge)	151.2	154.0
Steam at 1st receiver (absolute)	57.61	56.5
Steam at 2d receiver (absolute)	24.52	24.3
Vacuum in condenser in inches of mercury	24.3	24.67
Revolutions of main engines per minute	119.58	119.55
Piston speed in feet per minute	797.2	797.

		Wide.	Wide.
Throttle			
Air pressure in firerooms in inches of water		1.49	
Revolutions per minute of blowers		516.	
Temperatures in degrees Fahrenheit.	Fire rooms	117.6	
	Engine rooms	87.6	
	Injection	63.3	
	Discharge	113.3	111.4
	Feed	145.4	134.3
Revolutions of circulating pump		274.8	234.3
Fraction of stroke completed when steam was cut off.	H. P. cylinder	.6	.58
	I. P. cylinder	.72	.74
	L. P. cylinder	.76	.75
Mean Pressures	H. P. cylinder	61.959	57.727
	I. P. cylinder	21.299	19.913
	L. P. cylinder	8.912	9.986
Aggregate equivalent mean pressure on L. P. piston		30.51	30.1
I. H. P.	H. P. cylinder	1,645.105	1,560.47
	I. P. cylinder	1,352.175	1,325.24
	L. P. cylinder	1,249.73	1,400.46
Collective I. H. P., each main engine		4,247.01	4,286.17
Collective I. H. P., both main engines		8,533.18	
I. H. P. main circulating pumps		30.272	24.69
I. H. P. main air pumps		31.557	33.322
I. H. P. main air and circulating pumps		61.829	58.012
Collective I. H. P. main engines, air and circulating pumps		8,653.02	
I. H. P. of blowers		87.57	
I. H. P. of feed pumps		34.482	
I. H. P. of all other auxiliaries		39.696	
Aggregate I. H. P. of all machinery		8,814.79	
Cubic feet per minute swept by L. P. piston per I. H. P. main engines		7.54	7.47
Indicated thrust (I. H. P. of main engines only), pounds		57,481.	58,017.
Indicated thrust per square feet of helicoidal area of propellers (57.17 sq. ft.), pounds		1,006.7	1,016.1
Indicated thrust per square inch of thrust surface, pounds		47.66	48.11
Speed of ship, mean for 4 hours, knots		19.678	
Slip of propeller, per cent		18.2	18.2
$\frac{S^3 \times \text{immersed midship section}}{\text{I. H. P.}}$		716.67	
$\frac{S^3 \times D^2}{\text{I. H. P.}}$		232.89	
*Coal used per hour, pounds		26,354.43	
*Coal used per hour, per square foot of G. S		40.17	
*Coal used per hour, per I. H. P. (all machinery)		2.99	
Refuse per hour, pounds		1,713.037	

*(Approximate estimate from coal consumption trial.)

Refuse, per cent	6.5	
Velocity of steam through ports when wide open, in feet per second (mean piston speed):	Inboard.	Outboard.
H. P. steam port	107.6	149.3
I. P. steam port	126.8	175.9
L. P. steam port	172.2	235.2
H. P. exhaust port	83.4	
I. P. exhaust port	98.2	
L. P. exhaust port	134.7	
Main steam pipe	78.2	
Main exhaust pipe	108.9	
Total H. S. ÷ I. H. P	2.321	
I. H. P. per square foot of G. S. (all machinery)	14.126	
I. H. P. per ton of weight in engine compartments	29.477	
I. H. P. per ton of weight in boiler compartments	17.555	
I. H. P. per ton of weight propelling machinery and water	10.35	
I. H. P. per ton of weight all machinery, spare gear, tools and water	10.08	
Cooling surface per I. H. P.	1.567	1.555
Ratio of cooling surface to total H. S.	1 to	1.514
Wetted surface by Kirk's analysis, in square feet	20,100.	
I. H. P. per square foot of wetted surface at 19.678 knots	43.85	

MAXIMUM PERFORMANCE.

(Cards No. 9, taken during last ten minutes of trial trip.)	Starboard.	Port.
Steam at boilers (per gauge)	160.	
Steam at engines (per gauge)	155.	156.
Steam at 1st receiver (absolute)	60.2	59.
Steam at 2d receiver (absolute)	25.7	25.
Vacuum in inches of mercury	24.5	24.5
Throttle	Wide.	Wide.
Revolutions, main engines	120.8	120.26
Piston speed in feet per minute	805.33	801.73
I. H. P. of H. P. cylinder	1,660.85	1,569.46
I. H. P. of I. P. cylinder	1,433.35	1,365.50
I. H. P. of L. P. cylinder	1,306.64	1,404.06
Collective I. H. P., each main engine	4,400.84	4,339.02
Collective I. H. P., both main engines	8,739.86	
I. H. P. air and circulating pumps	61.826	58.012
I. H. P. main engine, air and circulating pumps	8,859.701	
I. H. P. blowers	87.59	
I. H. P. feed pumps	34.482	
I. H. P. all other auxiliaries	39.696	
I. H. P. all machinery	9,021.479	
Cubic feet per minute swept by L. P. piston per I. H. P. (main engines)	7.35	7.32

Air pressure in fire rooms in inches of water	1.54	
Cut-off in H. P. cylinder	.606	.586
Cut-off in I. P. cylinder	.719	.741
Cut-off in L. P. cylinder	.759	.75
I. H. P. per square foot of G. S.	14.46	
Total H. S. ÷ I. H. P	2.267	
I. H. P. per ton of propelling machinery and water	10.6	
I. H. P. per ton of all machinery, &c	10.3	
Speed (estimated for 15 minutes) in knots	19.66	
Slip of propeller per cent.	19.13	18.75

SYNOPSIS OF COAL CONSUMPTION TRIAL.

(Two main boilers in use under forced draft two hours.)

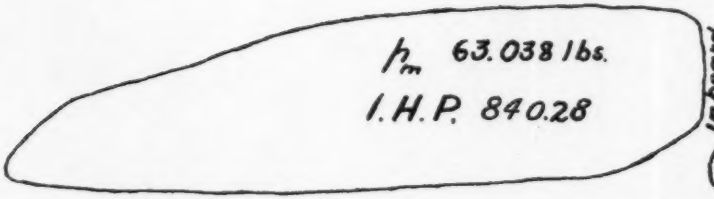
Steam at boilers (per gauge)	120.6	
Steam at engines (per gauge)	115.5	113.6
Steam at 1st receiver (absolute)	33.14	32.
Steam at 2d receiver (absolute)	14.25	11.87
Vacuum in condenser in inches of mercury	24.37	25.9
Throttle in holes (8 holes wide open)	2.	2¼
Revolutions, main engine	97.63	96.416
Piston speed in feet per minute	650.87	642.77
I. H. P. of H. P. cylinder	941.731	804.875
I. H. P. of I. P. cylinder	711.222	625.275
I. H. P. of L. P. cylinder	552.399	579.50
Collective I. H. P., each main engine	2,205.364	2,009.528
Collective I. H. P., main engine, air and circulating pump	2,234.211	2,037.18
Collective I. H. P., both engine, air and circulating pump	4,271.391	
Collective I. H. P., auxiliary machinery	109.695	
Collective I. H. P., total	4,381.086	
Cubic feet per minute swept by L. P. piston per I. H. P. (main engines)	11.86	12.86
Air pressure in fire rooms, in inches of water	1.62	
Cut-off in H. P. cylinder	.425	.409
Cut-off in I. P. cylinder	.719	.741
Cut-off in L. P. cylinder	.759	.75
I. H. P. per square foot of G. S.	14.04	
Total H. S. in use ÷ I. H. P	2.335	
Speed, in knots (taken by a Walker's "Cherub" log	17.55	
Slip of propeller	10.95	9.51
$\frac{S^2 \times \text{area of immersed amidship section}}{\text{I. H. P.}}$	1,031.38	
$\frac{S^3 \times \text{displacement } \frac{2}{3}}{\text{I. H. P.}}$	335.92	

The accompanying set of indicator diagrams are not corrected for indicator corrections, but the mean pressures and horse-powers are correct.

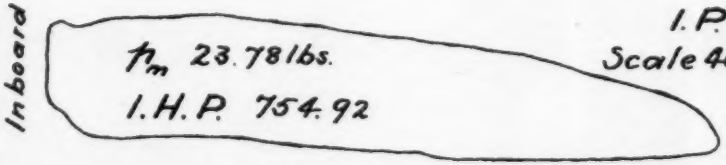
The diagram of valve motion is copied from the one furnished by contractor.

U. S. S. Philade

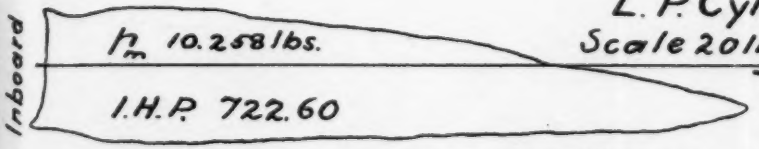
Starboard En



H. P. Cylinder
Scale 80 lbs. -



I. P. Cylin
Scale 40 lbs. - 1i

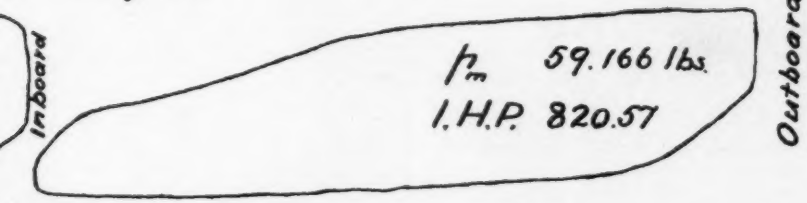


L. P. Cylinder
Scale 20 lbs - 1 inc

Steam at boilers 160 lbs (per gauge)
" " 155 " " "
" " 60.2 " (absolute)
" " 25.7 " "

iladelphia

d Engine



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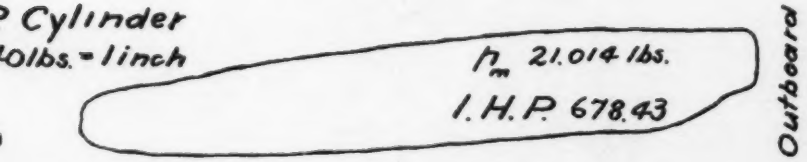

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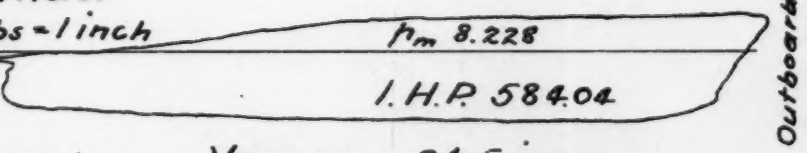

uge)
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Vacuum 24.5 ins.
Throttle wide
Revolutions 120.8
Total I.H.P. 4400.84 (main engine)

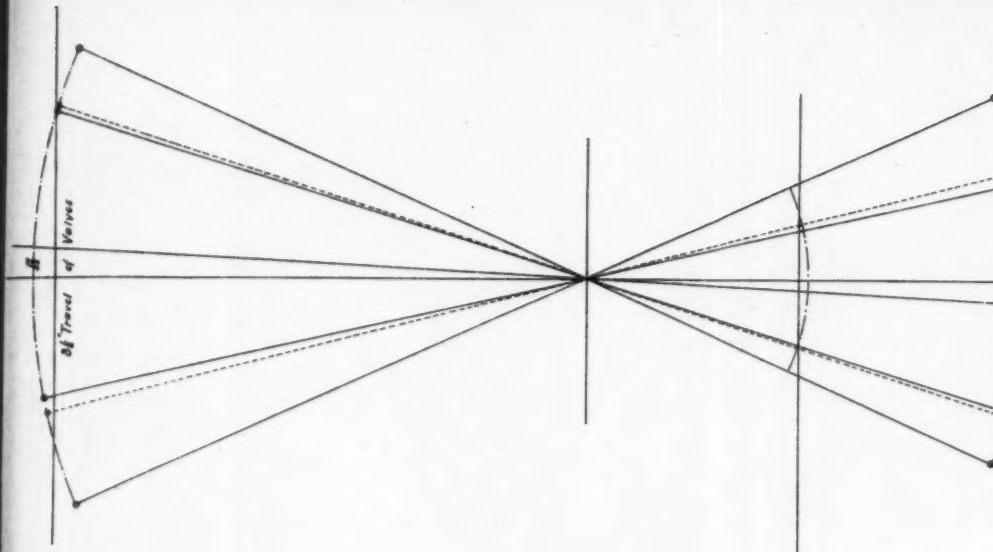
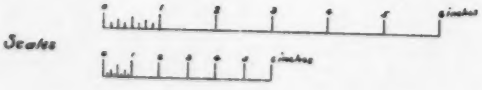
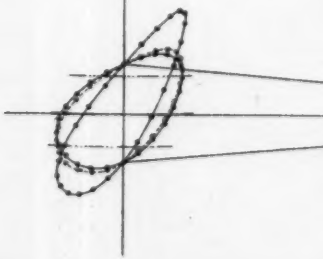


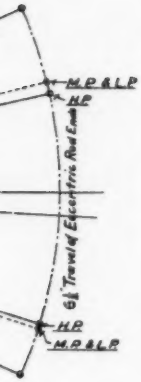
Diagram of Marshall Valve Gear

for the

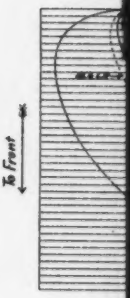
U.S.S. Philadelphia

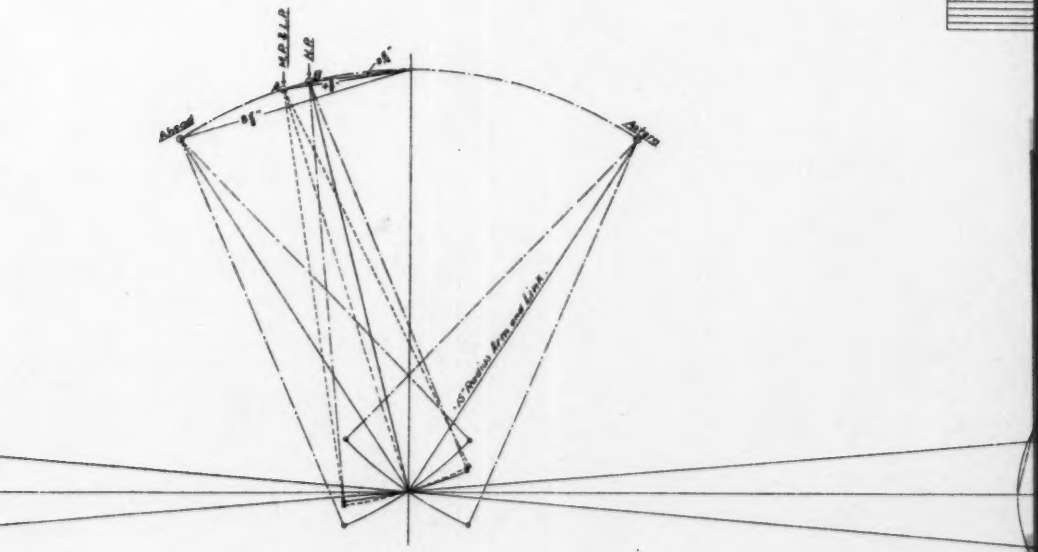


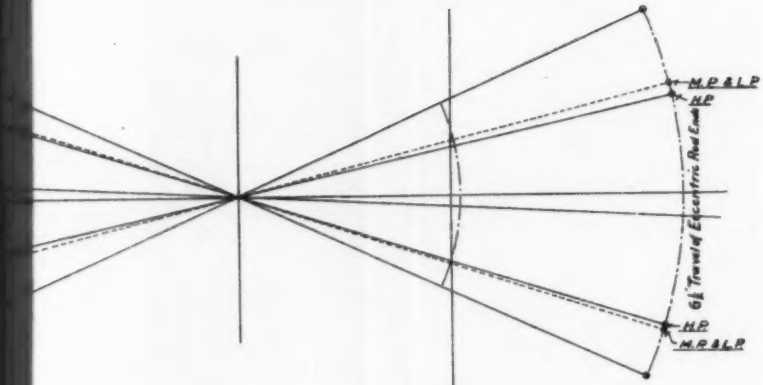




All Eccentrics with Cranks	Outboard Back End			Inboard Front End		
	H.P.	M.P.	L.P.	H.P.	M.P.	L.P.
Lap	1 1/16"	1 7/16"	1 1/16"	1 1/16"	1 1/16"	1 1/16"
Lead	1/4"	1/4"	1/4"	1/4"	1/4"	1/4"
Max. Opening	6 1/4" Forward front	6 1/4" Forward front	6 1/4" Forward front	6 1/4" Forward Back	6 1/4" Forward Back	6 1/4" Forward Back
Cut-off	31 1/8"	31 1/8"	32 1/8"	30 1/8"	30 1/8"	30 1/8"
Release	37 1/8"	37 1/8"	37 1/8"	35 1/8"	35 1/8"	35 1/8"
Cushioning	37 1/8"	37 1/8"	37 1/8"	35 1/8"	35 1/8"	35 1/8"
Average Cut-off at Full Gear H.P. .775 M.P. .775 L.P. .79						
Notching Up — H.P. A — M. & L.P. to A.						
Cut-off	1/8 Forward front	20 Forward front	2 1/4 Forward front	1 7/8 Forward Back	2 1/8 Forward Back	2 1/8 Forward Back
Average Cut-off H.P. 42 M.P. .507 L.P. .532						

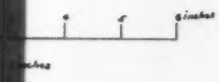


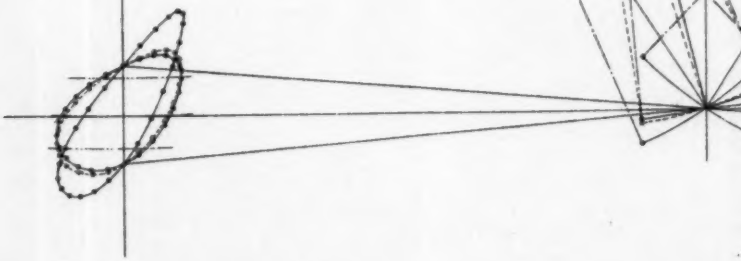




All Eccentr…		
	H.P.	
Lap	1 7/8"	
Lead	1/4"	
Max. Opening	4½ Forward	4½
Cut-off	31 5/8"	
Release	37 1/2"	
Cushioning	37 1/2"	
Average Cut-off at Fu…		
Notching Up —		
Cut-off	49 Forward	
Average Cut-off	H…	

…rshall Valve Gear
… the
…ladelphia



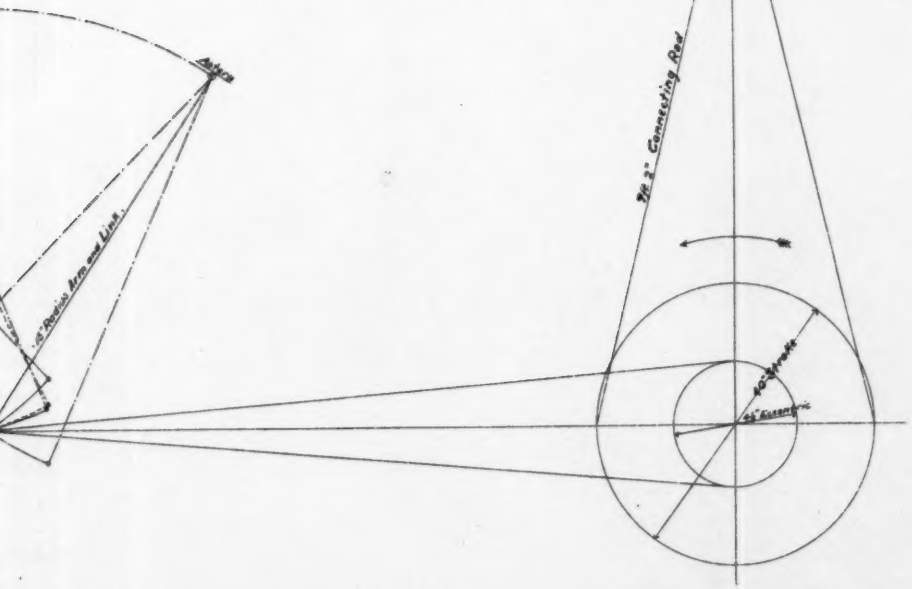


ntries with Cranks					
Outboard Back End			Inboard Front End		
	M.P.	L.P.	H.P.	M.P.	L.P.
	1⅞"	1⅞"	1⅞"	1⅞"	1⅞"
	⅝"	⅞"	⅝"	⅝"	⅝"
	44 Front	44 Front	36 Back	36 Back	36 Back
	31⅝"	32⅝"	30⅜"	30⅜"	30⅜"
	37⅜"	37⅜"	35⅝"	35⅝"	35⅝"
	37⅜"	37⅜"	35⅝"	35⅝"	35⅝"

Full Gear H.P. .715 M.P. .715 L.P. .79

	H.P. & B — M. & L.P. & A.					
20" Front	21½" Front	17½ Back	24" Back	21½" Back		

H.P. 82 M.P. .567 L.P. .532



XXIV.

CONTRACT TRIAL OF THE U. S. S. *SAN FRANCISCO.*

By Passed Assistant Engineer Edgar T. Warburton,
U. S. Navy.

The steel, twin-screw, schooner-rigged, protected cruiser *San Francisco* was built by the Union Iron Works, San Francisco, California.

She has a poop and a forecastle deck, with an open gun deck between. The protective deck is twenty-one inches above the water line amidships. It is two inches thick on the horizontal part, and three inches thick on the inclined sides. There are three steel masts, with sail area of 5,410 square feet. The fore and main masts have military tops.

The main battery consists of twelve 6-inch B. L. R., two mounted on the poop deck, two on the forecastle deck, and the others in sponsons on the gun deck. In the secondary battery are four 6-pounder R. F. G., four 3-pounder R. F. G., two 1-pounder R. F. G., three 37-mm. R.C. and four Gatlings. There are six above-water torpedo-launching tubes.

The contract called for an average speed of 19 knots per hour, to be maintained for four consecutive hours, with the ship loaded to a mean draught of 18¾ feet, a bonus of $50,000 being earned for each quarter knot over, and a penalty exacted of the same amount for each quarter knot under 19 knots.

HULL.

Length between perpendiculars	300 feet.
Length on L.W. L.	310 feet.
Length over all	324 feet 7 inches.
Beam, extreme	49 feet 1¼ inches.
Ratio of length to beam	6.308
Depth in hold from top of gun-deck beams to top of floors	28 feet 8 inches.
Draught, mean	18 feet 9 inches.

Displacement	4,088 tons.
Displacement per inch at L.W. L.	24.86 tons.
Volume of displacement	143,080 cubic feet.
Area of immersed midship section	770 square feet.
Area of L.W. L. plane	10,467 square feet.
Center of gravity of L.W. L. plane forward of midship section	4.943 feet.
Moment of inertia of L.W. L. plane	1,505,861.57
Center of buoyancy below L.W. L.	7.75 feet.
Center of buoyancy forward of midship section	0.743 feet.
Center of buoyancy forward of center between perpendiculars	0.72 feet.
Center of buoyancy below center of gravity	8.5 feet.
Center of gravity above L.W. L	0.75 feet.
Center of gravity above base line (assumed)	19.5 feet.
Transverse meta-center above L.W. L.	2.74 feet.
Transverse meta-center above C. B.	10.49 feet.
Transverse meta-center above C. G	1.99 feet.
Longitudinal meta-center above L.W. L	351.49 feet.
Longitudinal meta-center above C. B.	359.24 feet.
Longitudinal meta-center above C. G	350.74 feet.
Block coefficient of fineness	.509
Coefficient of fineness on midship section	.603
Coefficient of fineness of L.W. L.	.686
Coefficient of fineness of midship section	.832
Moment to alter trim one inch	399.93 foot tons.
Wetted surface, by Kirk's analysis	19,006 square feet.

MACHINERY.

It was originally intended to build the machinery of the *Baltimore* and *San Francisco* after the same plans, but a virtually new design for the *San Francisco* was the result of the alterations and modifications, made to conform to the plans prepared by Engineer-in-Chief Melville, before his appointment to that office, and these changes led to a considerable reduction in weight. The stroke was reduced from forty-two to thirty-six inches; the framing, pillow blocks and valve gear were altered; slide valves were retained only for the L. P. cylinders; independent air pumps were provided; the boilers were lengthened, and the furnaces and combustion chambers rearranged for the new system of forced draft, a substitution of closed ash pits for closed fire rooms.

A comparison of the following description of the machinery of the *San Francisco* as redesigned and built with that of the

Baltimore (pp. 337, 348, vol. I), will show the leading differences in the two designs.

Engines.—There are two horizontal, direct-acting, triple-expansion, screw engines placed in separate water-tight compartments; the engines are duplicates. The forward engine, which turns the port propeller, has the H. P. cylinder forward and the L. P. cylinder aft, this order being reversed in the after engine. The barrels and front heads of all cylinders and the L. P. covers are steam jacketed; reducing valves are fitted in the steam pipes of the I. P. and L. P. jackets, the working pressure being 135 pounds. There are two piston valves for each H. P. and I. P. cylinder and two double-ported, balanced slide valves for each L. P. cylinder; the H. P. valves take steam at the middle and the I. P. valves at the ends. The valve gear is of the Stephenson type, with double-bar links and rock shafts; the motion of the link block is transmitted to the rock shaft through a connecting link, one end of which is guided by a radius bar to reduce the slip of the block in forward gear. There are no cut-off valves, but each link can be independently adjusted, with a range of cut-off from 0.5 to 0.7 of the stroke, by means of a slotted link arm on the reversing shaft. All the links can be thrown in full backward gear without regard to the position of any of the suspension pin blocks in the slots. There is a starting valve for each cylinder, with lever on the working platform. The reversing engine has a horizontal steam cylinder with a hydraulic controlling cylinder; the steam valve is operated by a double floating lever and hand reversing lever; attached to the controlling cylinder is a double-acting plunger pump for reversing by hand. The throttle is a gridiron slide valve, operated through a lever and screw, with hand wheel. There is a separator in each main steam pipe close to the throttle valve.

The framing of the engine consists of forged steel struts, of which two connect each crank-shaft pillow-block to the corresponding cylinder; also two diagonal ties, one from the top of each L. P. pillow block to the valve rock-shaft bearings. The crank-shaft pillow-blocks are separate steel forgings, two for each section of crank shaft, bolted to a built-up steel bed-plate,

with provision for both horizontal and vertical adjustment. The H. P. and I. P. pistons are solid steel, dished castings. The L. P. piston is of manganese bronze with double shells. Each piston has two $\frac{5}{8}$-inch segmental packing rings of cast iron, the sections loosely doweled together. There is one interchangeable rod to each piston connecting with a forged steel crosshead, which has a composition slipper block fitted with a composition guide gib, the rubbing face of which is of white metal. The crosshead guide is of cast iron, hollow for water circulation, and with a forged steel backing guide bolted on each side.

The piston rods and connecting rods are forged of open-hearth steel. The crank pin and crank shaft brasses are lined with Parson's white metal. The crank pins are lubricated by telescopic oilers and wiper cups, and the crossheads by wiper oiling gear. Since the trial the eccentric straps have been lined with white-metal blocks.

	Starboard.	Port.
Number of cylinders (one engine)	3.	
Diameter of H. P. cylinder	$42\frac{1}{64}$ inches.	$42\frac{1}{64}$ inches.
Diameter of I. P. cylinder	$60\frac{9}{128}$ inches.	$60\frac{9}{128}$ inches.
Diameter of L. P. cylinder	$94\frac{11}{128}$ inches.	$94\frac{11}{128}$ inches.
Stroke of pistons	36 inches.	$36\frac{1}{32}$ inches.
Diameter of H. P. valves (two for each cylinder)	12 inches.	
Diameter of I. P. valves (two for each cylinder)	20 inches.	
Area of H. P. steam ports (both valves) for maximum opening.	Inboard ... 142.96 square inches.	
	Outboard.. 139. square inches.	
Area of H. P. exhaust ports (both valves) for maximum opening	204.5 square inches.	
Area of I. P. steam ports (both valves) for maximum opening.	Inboard ... 252.6 square inches.	
	Outboard.. 270.2 square inches.	
Area of I. P. exhaust ports (both valves) for maximum opening	423 square inches.	
Area of L. P. steam ports (both valves)	Inboard.... 489.5 square inches.	
	Outboard.. 486.7 square inches.	
Area of L. P. exhaust ports (both valves) for maximum opening	704 square inches.	
Area of main steam pipe	143.14 square inches.	
Area of H. P. exhaust pipe	226.98 square inches.	
Area of I. P. exhaust pipe	415.47 square inches.	
Area of L. P. exhaust pipe	706.86 square inches.	
	Starboard.	Port.
Volume swept by H. P. piston per stroke (net)	28.45 cubic feet.	28.52 cubic feet.

	Starbord.	Port.
Volume swept by I. P. piston per stroke (net)..	58.55 cubic feet.	58.60 cubic feet.
Volume swept by L. P. piston per stroke (net)..	144.46 cubic feet.	144.53 cubic feet.
Ratio of net area of H. P. piston to I. P. piston	2.057	2.0546
Ratio of net area of I. P. piston to L. P. piston	2.467	2.466
Ratio of net area of H. P. piston to L. P. piston	5.076	5.067
Clearance of H. P. cylinder, per cent	20.	
Clearance of H. P. cylinder, in inches of stroke	7.2	
Clearance of I. P. cylinder, per cent	22.85	
Clearance of I. P. cylinder, in inches of stroke	8.23	
Clearance of L. P. cylinder, per cent	11.64	
Clearance of L. P. cylinder, in inches of stroke	4.19	
Diameter of piston rods	$7\frac{1}{4}$ inches.	
Length of connecting rods between centers	72 inches.	
Diameters of connecting rods	$6\frac{1}{4}$ and $7\frac{1}{4}$ inches.	

Shafting and Thrust Bearings.—All the crank, line, thrust and propeller shafts are forged of open-hearth steel, with an axial hole through shafts and crank pins. The crank shafts are in three sections for each engine, all alike and interchangeable. When bolted together the cranks make angles of 120 degrees with each other, the H. P. leading, followed by the I. P., with the L. P. last for ahead motion.

The thrust and line shafts are separated by $\frac{3}{8}$-inch, the coupling bolts being loose in the thrust-shaft flange, and are lubricated by a centrifugal oiler. The propeller shafts are cased with composition in the stern tubes and bracket bearings. The thrust bearings are lined with white metal. The walls of the pedestal form an oil-trough, inside of which, both forward and abaft the thrust collars, are composition bearings to take the weight of the shaft. At each end of the bearing is a stuffing box to confine the oil.

Diameter of crank shafts	$14\frac{1}{4}$ inches.
Diameter of crank pins	15 inches.
Crank webs, thickness	$10\frac{1}{2}$ inches.
Crank webs, width	17 inches.
Diameter of line and thrust shafts	$13\frac{1}{4}$ inches.
Diameter of propeller shafts	$14\frac{1}{4}$ inches.
Diameter of hole in propeller shafts	9 inches.
Diameter of hole in other shafts and crank pins	7 inches.
Length of starboard shafting	102 feet $6\frac{3}{16}$ inches.
Length of port shafting	132 feet $4\frac{11}{16}$ inches.

Thrust collars, number (10 bearing)	11.
Thrust collars, diameters	14 and 18½ inches.
Thrust surface (actual)	1,135 square inches.

Condensers.—The condensers, one for each engine, are cylindrical, of cast brass, in four sections. The water circulates through the tubes.

Length over all	12 feet 9½ inches.
Diameter of shell inside	6 feet 6 inches.
Thickness of shell	½ inch.
Length of tubes exposed	9 feet $11\frac{5}{16}$ inches.
Diameter of tubes outside	⅝ inch.
Number of tubes (one condenser)	4,453.
Cooling surface (one condenser)	7,259 square feet.
Cooling surface of main condensers to H. S. of main boilers	1 to 1.354.
Cooling surface of main condensers to H. S. of main and auxiliary boilers	1 to 1.386.

Auxiliary Condenser.—In the forward engine room is a vertical, cast brass, auxiliary condenser of rectangular section. The pumps are operated by a two-cylinder engine. The feed pump delivers into a tank in the auxiliary boiler room on the gun deck. The auxiliary condenser can be connected with the main condensers. Cooling surface, 294.5 square feet.

Air and Bilge Pumps.—The air pumps are horizontal, single-acting, two for each main engine driven by a vertical compound engine.

The engines are connected to the pumps by cast-iron frames, which form the pump crosshead guides. The pump rods are connected to the same crank pins as are the engine connecting rods. Each air pump drives, through the prolonged pump rods, two horizontal plunger bilge pumps.

Diameter of H. P. cylinder	12 inches.
Diameter of L. P. cylinder	19 inches.
Diameter of air pumps	26 inches.
Diameter of bilge pumps	6 inches.
Stroke, common	18 inches.
Ratio of net volume of main L. P. cylinder to net volume of air pumps	13.21 to 1.

Circulating Pumps.—There is a centrifugal circulating pump of composition for each condenser driven by a vertical, inverted, balanced, compound engine. The pump has suction from the sea and bilge, and delivery to the condenser and overboard.

Diameter of H. P. cylinder	7½ inches.
Diameter of L. P. cylinder	12 inches.
Stroke	6 inches.
Diameter of runner	36 inches.
Width of opening in rim of runner	3 inches.
Capacity per minnte	8,000 gallons.

Screw Propellers.—The propellers are of the modified Griffiths' type, with adjustable blades, of manganese bronze. The starboard propeller is right-handed and the port propeller left-handed. The hubs are spherical, with conical tail piece.

Number of blades		3.
Diameter		13 feet 6 inches.
Pitch, limits of adjustment:		
Entering edge		17 feet and 19 feet.
Leaving edge		18 feet 6 inches and 20 feet 6 inches.
Pitch, as set on trial:		
Entering edge	18 feet.	Mean 18 feet 9 inches.
Leaving edge	19 feet 6 inches.	
Greatest width of blade at 3 feet 3 inches radius		4 feet 7¾ inches.
Helicoidal area		57.59 square feet.
Projected area		44.55 square feet.
Disc area		143.14 square feet.
Pitch ÷ diameter		1.39
Helicoidal area ÷ disc area		.402
Projected area ÷ disc area		.311
Disc area of both propellers ÷ area of immersed midship section		.372
Immersion of center of propeller at draught of 20 feet 6 inches aft		9 feet 1¼ inches.
Diameter of hub		3 feet 11 inches.

Main Boilers.—There are four steel, cylindrical, double-ended, horizontal, fire-tube boilers placed fore and aft in two watertight compartments, with four athwartship fire rooms. There are three corrugated furnaces at each end of a boiler, and each has a separate combustion chamber.

The material of plates, furnaces, braces and rivets, is open-hearth steel; the tubes and grate bars are wrought iron. The longitudinal joints of shells are butted, with inside and outside straps and treble-riveted. The circumferential joints are lapped and double-riveted. There are two standing oval smoke pipes, one for each pair of boilers, surrounded by casings which leave an annular space of five inches.

On each boiler are two Weir hydrokineters for circulating water in raising steam.

Length of boiler	19 feet 2 inches.
Diameter	14 feet 8 inches.
Number of furnaces in each	6.
Inside diameters of furnaces	42 and 45 inches.
Length of grate	6 feet 7 inches.
Number of common tubes	832.
Thickness	No. 12 B. W. G.
Number of stay tubes	224.
Thickness	No. 6 B. W. G.
Total number of tubes	1,056.
Diameter of tubes outside	2¼ inches.
Length between tube sheets	6 feet 9 inches.
Depth of combustion chambers	2 feet 5 inches.
Width of middle combustion chamber	3 feet 2½ inches.
Height of middle combustion chamber above grate	8 feet 1 inch.
Width of wing combustion chambers	4 feet.
Height of wing combustion chambers above grate	6 feet 5 inches.
Thickness of shell plates	1 1/32 inches.
Thickness of heads: upper plate	1 1/32 inches.
Thickness of heads: middle plate	¾ inch.
Thickness of heads: lower plate	11/16 inch.
Thickness of tube sheets	¾ inch.
Thickness of furnaces	½ inch.
Thickness of combustion chamber plates	½ inch.
Diameter of rivets for longitudinal joints	1 1/16 inches.
Diameter of rivets for circumferential joints	1 inch.
Number of through braces above the tubes	9.
Diameters	2¼ and 2½ inches.
Number of through braces above lower man holes	2.
Number of diagonal braces	8.
Number of braces from heads to back tube sheets	4.
Diameters of these braces	1 3/16 inches.
Tube heating surface	4,198.75 square feet.
Plate heating surface	715.64 square feet.
Total heating surface	4,914.39 square feet.
Grate surface	138.25 square feet.
Area through tubes	22.89 square feet.
Water surface, 8 inches above upper tubes	240 square feet.
Steam room, 8 inches above upper tubes	672.4 cubic feet.
Capacity of combustion chambers and furnaces above grate	576 cubic feet.
Diameters of after smoke pipe	7 feet 7 inches by 5 feet 5 inches.
Area of after smoke pipe	34.8 square feet.
Diameters of forward smoke pipe	7 feet 7 inches by 6 feet.

Area of forward smoke pipe (net)..... 33 square feet.
Height of smoke pipes above lower grates..... 55 feet.
Load on safety valve..... 138 lbs.
Test pressure..... 225 lbs.

TOTALS FOR THE FOUR MAIN BOILERS.

Tube heating surface..... 16,795.00 square feet.
Plate heating surface..... 2,862.56 square feet.
Total heating surface..... 19,657.56 square feet.
Grate surface..... 553 square feet.
Area through tubes..... 91.56 square feet.
Area of smoke pipes..... 67.8 square feet.
Area of water surface..... 960 square feet.
Capacity of steam room..... 2,689.6 cubic feet.
Capacity of combustion chambers and furnaces above grates..... 2,304 cubic feet.
Ratios:
Tube H. S. to G. S..... 30.37 to 1.
Plate H. S. to G. S..... 5.18 to 1.
Total H. S. to G. S..... 35.55 to 1.
Area through tubes to G. S..... .165 to 1.
Area of smoke pipes to G. S..... .122 to 1.
Area of water surface to G. S..... 1.74 to 1.
Steam room per square foot of G. S..... 4.86 cubic feet.
Capacity of combustion chambers and furnaces per square foot of G. S..... 4.16 cubic feet.
Weight of boilers, fittings, water and all fire room weights, per 100 square feet H. S., in tons..... 2.279.

Auxiliary Boiler.—In a compartment on the gun deck is an auxiliary steel, cylindrical, single-ended, horizontal, return fire-tube boiler, with one corrugated furnace fitted with ash-pit draft similar to the main boiler system. The smoke pipe is inside the forward main smoke pipe.

Length..... 8 feet.
Diameter..... 8 feet.
Inside diameters of furnace..... 39 and 42 inches.
Length of grate..... 4 feet 5 inches.
Number of common tubes..... 82.
Thickness..... No. 12 B. W. G.
Number of stay tubes..... 32.
Thickness..... No. 6 B. W. G.
Diameter of tubes outside..... 2¼ inches.
Length between tube sheets..... 5 feet 7⅛ inches.
Tube heating surface..... 375.5 square feet.
Plate heating surface..... 100.7 square feet.

Total heating surface	476.2 square feet.
Grate surface	14.6 square feet.
Area through tubes	2.44 square feet.
Water surface, 8 inches above upper tubes	54 square feet.
Steam room, 8 inches above upper tubes	79.9 cubic feet.
Capacity of combustion chamber and furnace above grate	58.3 cubic feet.
Diameter of smoke pipe	21 inches.
Area of smoke pipe	2.4 square feet.
Load on safety valves	138 pounds.
Test pressure	225 pounds.

TOTALS FOR THE MAIN AND AUXILIARY BOILERS.

Tube heating surface	17,170.50 square feet.
Plate heating surface	2,963.26 square feet.
Total heating surface	20,133.76 square feet.
Grate surface	567.6 square feet.
Area through tubes	94 square feet.
Area of smoke pipes	70.2 square feet.
Area of water surface	1,014 square feet.
Capacity of steam room	2,769.5 cubic feet.
Capacity of combustion chambers and furnaces above grates	2,362.3 cubic feet.

RATIOS OF MAIN AND AUXILIARY BOILERS.

Tube H. S. to G. S	30.25 to 1.
Plate H. S. to G. S	5.22 to 1.
Total H. S. to G. S	35.47 to 1.
Area through tubes to G. S	.166 to 1.
Area of smoke pipes to G. S	.123 to 1.
Area of water surface to G. S	1.78 to 1.
Steam room per square foot of G. S	4.88 cubic feet.
Capacity of combustion chambers and furnaces per square foot of G. S.,	4.16 cubic feet.
Weight of main and auxiliary boilers, fittings, water and all fire room weights per 100 square feet H. S., in tons	2.309

Forced Draft.—The system consists of blowers discharging into the closed ash-pits. There are two blowers, driven by enclosed compound engines, in each fire room. The air ducts from the blowers unite above and below the floors, and the admission of air into the ash-pits is controlled and regulated by a damper for each ash-pit, there being also a damper close to each blower. Openings in the furnace fronts regulated by dampers supply air to the furnaces from the ash-pits.

Main blowers:

Number	8.
Diameter of H. P. cylinder	5 inches.

Diameter of L. P. cylinder	8 inches.
Stroke	6 inches.
Diameter of fan	60 inches.
Width of opening in rim of fan	7½ inches.
Auxiliary boiler-blower engine	4 × 4 inches.
Diameter of fan	40 inches.
Width of opening in rim of fan	3¾ inches.

Feed Pumps.—In the after fire room of each boiler compartment are two vertical, double, double-acting fly-wheel pumps for main and auxiliary feed, four in all.

Diameter of steam cylinders (2)	11 inches.
Diameter of water cylinders (2)	6½ inches.
Stroke	10 inches.
Auxiliary boiler feed pump	Dow No. 3.
Steam and water cylinders	4½ and 2¾ inches.
Stroke	6 inches.

Bunkers.—There are fourteen bunkers in the hold and thirty-six in double rows on the sides of the ship, on the protective deck, fifty-four bunkers in all.

Capacity of feeding bunkers	242.16 tons.
Capacity of wing bunkers on protective deck	301.4 tons.
Capacity of middle line bunkers on protective deck	84 2 tons.
Total bunker capacity	627.76 tons.
Coal supply at mean draught of vessel, 18 feet 9 inches	325 tons.

Other Auxiliary Machinery.—Two double double-acting yoke fly-wheel pumps, 8½ × 6¼ × 10 inches, for bilge, fire and water service purposes, one in each engine room.

Two double-engines, 6 × 6 inches, for turning the main engines, one in each engine room.

Four double ash-hoists, 4 × 6 inches, one for each fire room.

A three-cylinder steering engine, 10 × 6 inches, under the protective deck aft.

A two-cylinder, reversible windlass, 10 × 20 inches, with two wildcats.

Two Sturtevant blowers, 6 × 6 inches, with 54-inch fans, for ventilating purposes.

Distilling Apparatus.—Two Baird double-coil distillers with filters, and two Baird evaporators; capacity, 5,000 gallons per twenty-four hours.

One Davidson pump, $5\frac{1}{2} \times 6 \times 10$ inches, for circulating water through the distillers and for flushing purposes. One Davidson pump, $4\frac{1}{2} \times 2\frac{1}{2} \times 6$ inches, for feeding the evaporators.

Electric Light Plant.—There are three Edison compound, multipolar, marine dynamos of 80 volts, 200 ampères, 400 revolutions; with 423 incandescent lamps of 5,762 candle-power; also four Mangin projectors with six mounts, manufactured by Sautter, Lemonnier et Cie, Paris. Each dynamo is directly coupled to an inverted, inclined, compound engine; diameters, $7\frac{1}{2}$ and 12 inches; stroke, 6 inches.

WEIGHTS.

Propelling machinery	740.31 tons.
Water in above machinery	140.75 tons.
Tools, instruments and duplicate parts	24.24 tons.
Miscellaneous machinery	8.82 tons.
Total weight at time of completion	914.12 tons.
One main boiler with fittings, empty	59.85 tons.
Water in one main boiler	31.38 tons.
All main boilers with fittings and water	364.92 tons.
All main boilers, fittings, water, uptakes, smoke pipes, and all machinery in fire rooms	448.16 tons.
Auxiliary boiler with fittings, empty	9.68 tons.
Water in auxiliary boiler	4.7 tons.
Auxiliary boiler, fittings, water, uptake, smoke pipe, and all machinery in fire room	16.83 tons.
Total of main and auxiliary boilers, fittings, water, uptakes, smoke pipes, and all machinery in fire rooms	464.99 tons.

Reciprocating parts:	Starboard.	Port.
H. P	7,612 lbs.	7,615 lbs.
I. P	8,875 lbs.	8,963 lbs.
L. P	12,192 lbs.	12,265 lbs.

TRIAL.

Leaving San Francisco August 25th, 1890, the run to Santa Barbara was made without incident at a speed of about 13 knots per hour.

The trial course, measuring 74,113.1 meters, was laid off in Santa Barbara Channel, about one mile from the prominent points of the shore line, the eastern end about four miles east of Santa Barbara Light and the western end near Point Concepcion.

On the 27th August, in the forenoon, this course was run over twice in opposite directions. The elapsed time was 4 hours 5 minutes 57.375 seconds. There was a current of .459 knots per hour setting to the westward. On the return run the course was increased in length by the sweep made in getting back and the steering.

At the start the throttle valves were half open. At the beginning of the last hour the opening was increased to three-quarters. This was the only change in the adjustment of the machinery during the trial, the various cut-offs being as recorded. Steam was on the L. P. jackets. The engines moved with great steadiness. The centers were turned smoothly without jar. The vibration was insignificant.

A very moderate amount of water was used on the crank-shaft journals, crank pins and crossheads. Water circulated through the hollow crosshead slides. The only exception to the perfect working of the machinery was the heating and wearing of the ahead eccentric straps, which required much water. The accumulation of water in the engine rooms from the water service drained into the fire rooms through an open valve which was supposed to be closed, and then into the air-ducts, making it necessary to shut off the draft from six furnaces for nearly half an hour, and from two others during the whole return run. For more than an hour spray was blown into the ash-pits of three other furnaces. There was a drop in the steam pressure of nearly ten pounds through this accident, but the good work of the firemen regained the loss, and more steam was made during the last hour than at any other period. The loss in time, which was through no fault of the engines or boilers, was calculated from the reduced engine speed to be 1 minute 42.873 seconds.

At the beginning of the trial steam was furnished by the main boilers, the auxiliary boiler being connected an hour later. There was no foaming in the boilers. Fresh water was used.

The trial showed the effectiveness of the closed ash-pit system of forced draft. Harris Navigation, hand picked, semi-bituminous coal (Welsh) was burned with little smoke and a small amount of refuse. The same coal was used on the

Charleston's trial, with closed fire rooms, when great volumes of smoke issued from the smoke pipe and the shower of cinders on deck was very much greater than on the *San Francisco.* Again, the *Charleston's* up-takes and smoke pipe at the base were at times red hot, but in the *San Francisco,* though the combined area of the two smoke pipes is smaller than the *Charleston's,* there was no overheating and the paint was not blistered. The fire rooms were comparatively comfortable and free from the clouds of dust which pervade air-tight fire rooms.

No tubes leaked at the front end, and only eight to a slight extent at the beaded end in the combustion chamber; six of these were in the same nest in a boiler that was under forced draft eight hours longer than any other. The few leaking tubes were the result of the whole trip to and from Santa Barbara, as ascertained on the return to the Works.

Indicator cards were taken every half hour from the main engines; those selected for illustration being average cards.

The mean draught of the vessel on the trial was exactly that required by the contract; pig iron was carried to represent the weights of armament, masts and stores.

Throughout the trial the sea was smooth; on the first run the wind was on the bow with force from 1 to 2; on the return run the wind was on the quarter and the force was 3.

The trial crew, under Chief Engineer Robert Forsyth, was composed of 77 engineers, machinists, oilers, water tenders and storekeepers and 50 firemen.

On the return run to San Francisco the speed of the ship being 13½ knots, with the helm hard-a-starboard, a complete circle was turned in 5 minutes 32 seconds. With helm hard-a-port, port engine going ahead and starboard engine backing, a circle was completed in 6 minutes 2 seconds; with helm hard-a-starboard, starboard engine going ahead and port engine backing, a circle was completed in 6 minutes 12 seconds. Steering by hand with helm hard-a-starboard, a circle was made in 7 minutes 41 seconds.

DATA OF TRIAL.

Draught at beginning { forward..... 17 feet 4 inches / aft..... 20 feet 4 inches } ..... 18 feet 10 inches.

Draught at end { forward..... 16 feet 10 inches / aft ..... 20 feet 6 inches } ..... 18 feet 8 inches.

Draught mean for trial { forward..... 17 feet 1 inch / aft..... 20 feet 5 inches } ..... 18 feet 9 inches.

Displacement for mean draft..... 4,088 tons.
Area of immersed midship section..... 770 square feet.
Wetted surface, Kirk's analysis..... 19,006 square feet.
I. H. P. per 100 square feet of wetted surface..... 50.41
I. H. P. per 100 square feet of wetted surface at 10 knots in ratio of 3.5 power...4.85
Average speed corrected for steering..... 19.525 knots.
Average speed due to elapsed times over course..... 19.518 knots.
Slip, for 19.525 knots (mean of both screws)..... 15.45 per cent.
Speed over last 7,232 meters of course between bearings..... 20.115 knots.
Average speed for whole run by patent log..... 20.604 knots.
$Speed^3 \times$ area immersed midship section $\div$ I. H. P..... 598.72
$Speed^3 \times$ displacement $^{\frac{2}{3}} \div$ I. H. P..... 198.63

SYNOPSIS OF STEAM LOG.

		Starboard.	Port.
Revolutions of main engine per minute		125.798	123.835
Revolutions, mean of both engines		124.817	
Piston speed, feet per minute		754.78	743.65
Steam at boilers (gauge)		135.3	
Steam at engines, (gauge)		126.2	128.4
Steam at 1st receiver (absolute)		55.1	58.1
Steam at 2d receiver (absolute)		19.25	18.25
Vacuum in condenser in inches of mercury		25.7	26.1
Opening of throttle		.55	
Steam cut-off in fraction of stroke from the beginning.	H. P.	.69	.69
	I. P.	.69	.69
	L. P.	.69	.70
Double strokes of air pump per minute		197.	131.
Revolutions of circulating pump per minute		199.7	201.25
Temperatures in degrees Fahrenheit.	Steam at H. P. cylinder	341.4	343.
	Engine rooms	100.7	94.8
	Injection	63.7	
	Discharge	101.2	99.3
	Feed	109.5	103.1
	Fire rooms	114.1	
Revolutions of blowers per minute		478.	
Air pressure in ducts in inches of water		2.02	
H. P. cylinder	Mean pressure	51.37	46.45
	I. H. P.	1,590.06	1,427.92
I. P. cylinder	Mean pressure	25.34	26.79
	I. H. P.	1,603.63	1,700.43

	Starboard.	Port.
L. P. cylinder { Mean presssure	9.93	10.64
L. P. cylinder { I. H. P.	1,594.39	1,664.29
Aggregate equivalent mean pressure on L. P. piston	30.32	30.68
Collective I. H. P. each main engine	4,788.08	4,792.64
Collective I. H. P. both main engines	9,580.72	
I. H. P. of air pumps	66.48	41.23
I. H. P. of circulating pumps	14.36	15.07
Collective I. H. P. air and circulating pumps	137.14	
I. H. P. main blowers	110.67	
I. H. P. main feed pumps	42.85	
I. H. P. other auxiliaries	41.55	
Collective I. H. P. all auxiliaries	332.21	
Collective I. H. P. each main engine, air and circulating pumps	4,868.92	4,848.94
Collective I. H. P. both main engines, air and circulating pumps	9,717.86	
Total I. H. P. main engines and all auxiliary machinery in operation	9,912 93	
Indicated thrust (I. H. P. main engines)	66,988.51 lbs.	68,115.38 lbs.
Indicated thrust per square foot of developed area of propeller (57.59 square feet)	1,163.19 lbs.	1,182.76 lbs.
Indicated thrust per square inch of surface of thrust bearing	59.02 lbs.	60.01 lbs.
Cubic feet swept per minute by L. P. piston per I. H. P.	7.59	7.47
Square feet of cooling surface per I. H. P.	1.48	1.44
Squre feet of total heating surface per I. H. P.	2.03	
I. H. P., all machinery in operation, per square foot of G. S.	17.46	
I. H. P. per ton of main and auxiliary boilers, water, fittings and all fire-room weights	21.32	
I. H. P. per ton of propelling machinery, boilers and water	11.25	
I. H. P. per ton of total machinery	10.84	
Velocity of steam, in feet per second, through the ports at the maximum opening of the valves:		
H. P. steam port	117.52	124.60
I. P. steam port	137.82	130.80
L. P. steam port	176.26	178.01
H. P. exhaust port	83.38	
I. P. exhaust port	85.35	
L. P. exhaust port	126.46	
Main steam pipe	118.92	
Main exhaust pipe	125.97	

MAXIMUM PERFORMANCE.

Card No. 9 (taken in last half hour of trial).

Revolutions of main engines per minute	127.8	126.33
Revolutions, mean of both engines	127.065	

		Starboard.	Port.
Piston speed, feet per minute		766.80	758.64
Steam at boilers (gauge)		142.5	
Steam at engines (gauge)		132.	137.
Steam at 1st receiver (absolute)		59.7	65.7
Steam at 2d receiver (absolute)		20.7	19.2
Vacuum in condenser in inches of mercury		25.5	25.75
Opening of throttle		.75	
Steam cut off in fraction of stroke from beginning.	H. P.	.69	.69
	I. P.	.69	.69
	L. P.	.69	.70
Double strokes of air pump per minute		216.	134.
Revolutions of circulating pump per minute		201.	190.
Temperatures in degrees Fahrenheit.	Steam at H. P. cylinder	345.	342.
	Engine room	102.	95.
	Injection	65.	
	Discharge	105.	100.
	Feed	106.	106.
	Fire room	113.5	
Revolutions of blowers per minute		495.	
Air pressure in duct in inches of water		2.5	
H. P. cylinder	Mean pressure	49.50	47.47
	I. H. P.	1,571.09	1,493.17
I. P. cylinder	Mean pressure	27.37	29.65
	I. H. P.	1,788.25	1,915.10
L. P. cylinder	Mean pressure	10.51	11.34
	I. H. P.	1,694.51	1,806.59
Aggregate equivalent mean pressure on L. P. piston		31.35	32.72
Collective I. H. P., each main engine		5,053.85	5,214.86
Collective I. H. P., both main engines		10,268.71	
I. H. P. of air and circulating pumps		81.39	59.15
Collective I. H. P., air and circulating pumps		140.54	
I. H. P., other auxiliaries		195.07	
Collective I. H. P., all auxiliaries		335.61	
Collective I. H. P., each main engine, air and circulating pumps		5,135.24	5,274.01
Collective I. H. P., both main engines, air and circulating pumps		10,409.25	
Total I. H. P., main engines and all auxiliary machinery in operation		10,604.32	
Cubic feet swept per minute by L. P. piston per I. H. P.		7.3	7.0
Square feet of total H. S. per total I. H. P.		1.89	
I. H. P., all machinery in operation per square foot G. S.		18.68	
I. H. P. per ton of main and auxiliary boilers, fittings, water and all fire-room weights		22.80	
I. H. P. per ton of propelling machinery, boilers and water		12.03	
I. H. P. per ton of total machinery		11.60	

It would be very interesting to compare the performance of the *San Francisco* with those of the *Baltimore* and the *Philadelphia* on account of the similarity of the hulls, though slightly different in size, and the fact that the machinery of all three ships was designed for the same horse-power. Unfortunately, the *Baltimore's* speed was only obtained by patent log, and the accident by water in the air ducts, coupled with the fact that the firemen did not exert themselves as they should have done during the first run, makes the *San Francisco's* speed and power undoubtedly lower than she can do under the best circumstances. The best proof of the excellent record which the latter vessel can make is shown by the speed made just at the end of the course, when the performance is usually below the maximum. For a distance of 7,232 meters (about 3.9 miles) the speed was 20.11 knots without allowing for the current, which was adverse. I am entirely satisfied that with the eccentric straps as now lined with white metal, the *San Francisco* is good for 20 knots for four hours.

A few words in regard to the closed ash-pit forced draft, in addition to what I have already said, may not be amiss. It worked without a hitch, and the firemen seemed entirely "at home" with it, which they never do with the closed fire rooms. As was frequently remarked, it was as easy to shut off and regulate as a throttle.

The great advantages of this plan are the perfect regulation, the greater comfort of the fire rooms, and confidence of the people working in them, and, above all, the better treatment of the boilers. When firing or cleaning a fire there is no inrush of cold air under pressure, and consequently an almost total elimination of the danger of leaky tubes and seams. This was shown conclusively on this trial, where only six tubes showed a little weeping in a boiler which had been under an air pressure of two inches for about thirteen hours.

I regret to observe in the recent specifications of machinery for our new vessels that this system is not continued, but I am informed that it is due to the difficulty of locating the blowers. In order to ventilate the fire rooms properly it is necessary to

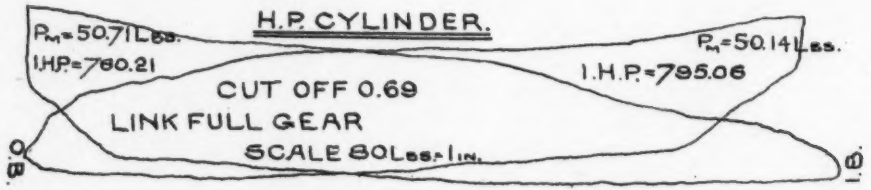
U.S.S. San Francisco.

Stbd. Main Engines Cards No 5.

9.40 A.M. August 27th 1880.

STEAM PRESSURE IN BOILERS 137 POUNDS.
" " AT ENGINES 125 POUNDS.
VACUUM IN INCHES 26.
THROTTLE VALVE 4.
REVOLUTIONS PER MINUTE 125.8

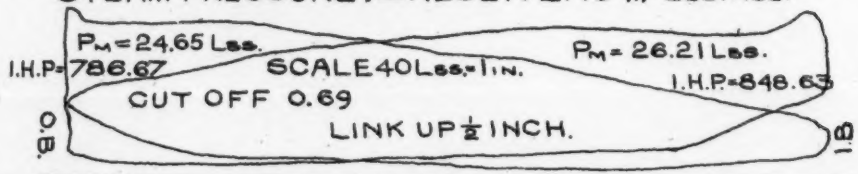
H.P. CYLINDER.



I.P. CYLINDER

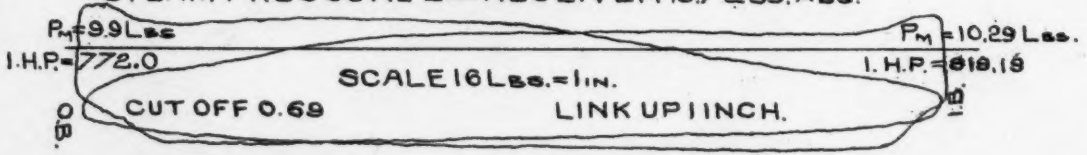
I.P. CYLINDER

STEAM PRESSURE 1ST RECEIVER 54.7 Lbs. ABS.



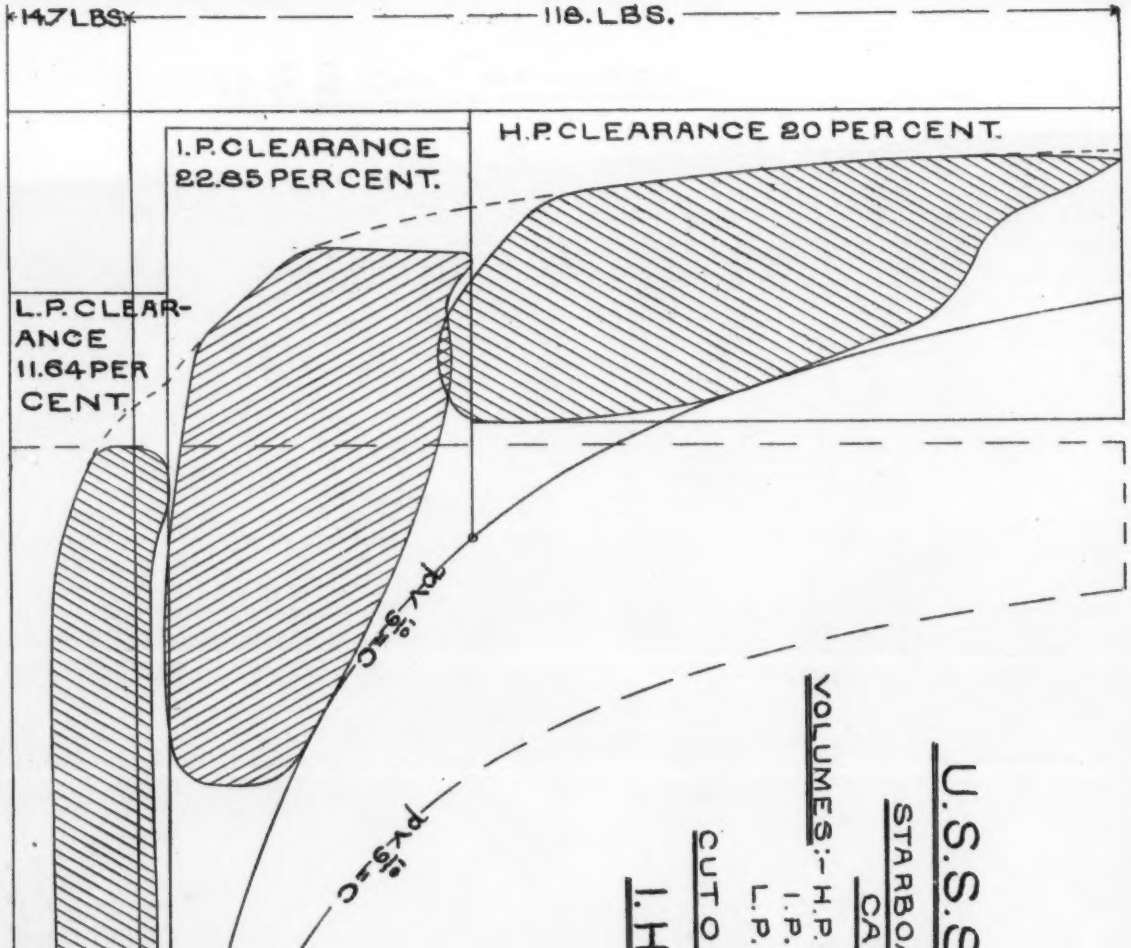
L.P. CYLINDER

STEAM PRESSURE 2ND RECEIVER 19.7 Lbs. ABS.



TEMPERATURE OF INJECTION 61 DEGREES.
" " DISCHARGE 107 "
" " FEED 96 "

TOTAL I.H.P. = 4801.78


147 LBS.
118. LBS.
I.P. CLEARANCE 22.85 PER CENT.
H.P. CLEARANCE 20 PER CENT.
L.P. CLEARANCE 11.64 PER CENT.
PV^1.3=C
PV^1.3=C
U.S.S.
STARBOA
CA
VOLUMES:- H.P.
I.P.
L.P.
CUT O
I. H

S.S. SAN FRANCISCO.

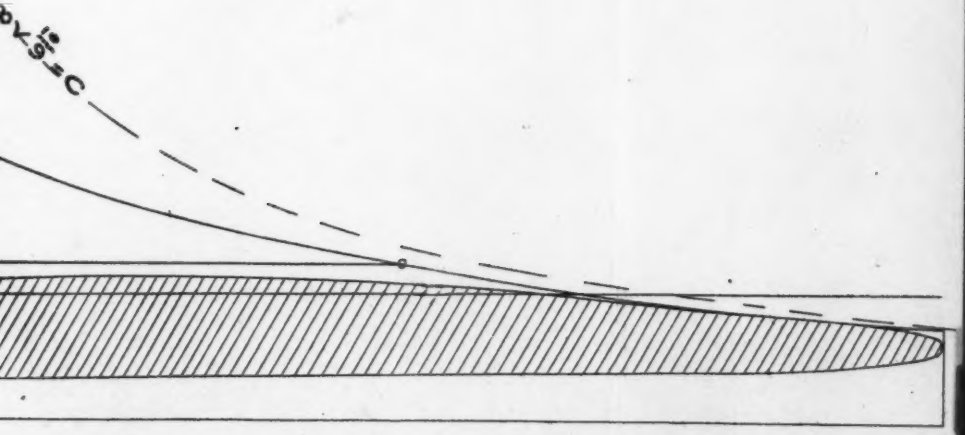
STARBOARD ENGINES

CARDS No. 5

S:- H.P. CYLINDER 28.45 CUBIC FEET.
I.P. ,, 58.55 ,, ,,
L.P. ,, 144.46 ,, ,,

CUT OFF H.P. CYLINDER 0.69

I.H.P. 4801.786



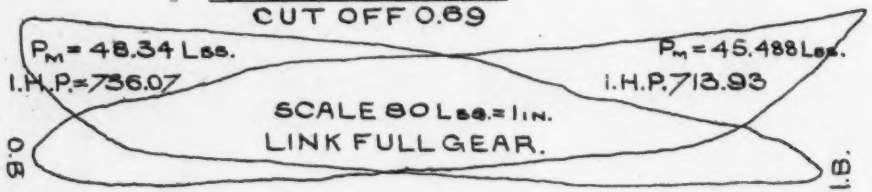
U. S. S. San Francisco.

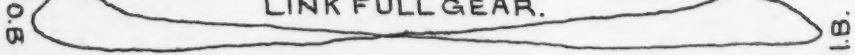
Port Main Engines Cards № 5.

9.40 A.M. August 27th 1890.

STEAM PRESSURE IN BOILERS 137 POUNDS
" " AT ENGINES 130 POUNDS
VACUUM IN INCHES 26.5
THROTTLE VALVE 4.
REVOLUTIONS PER MINUTE 124.21

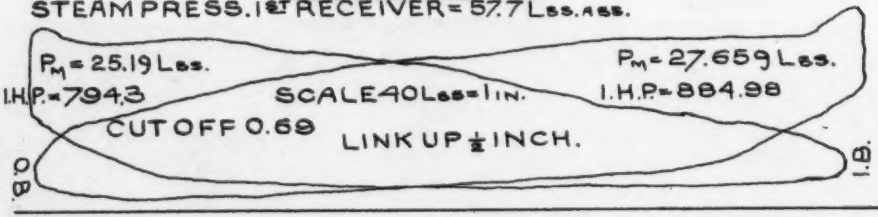
H.P. CYLINDER
CUT OFF 0.69





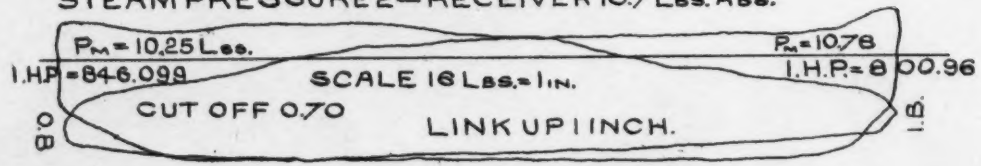
I.P. CYLINDER

STEAM PRESS. 1ST RECEIVER = 57.7 Lbs. ABS.

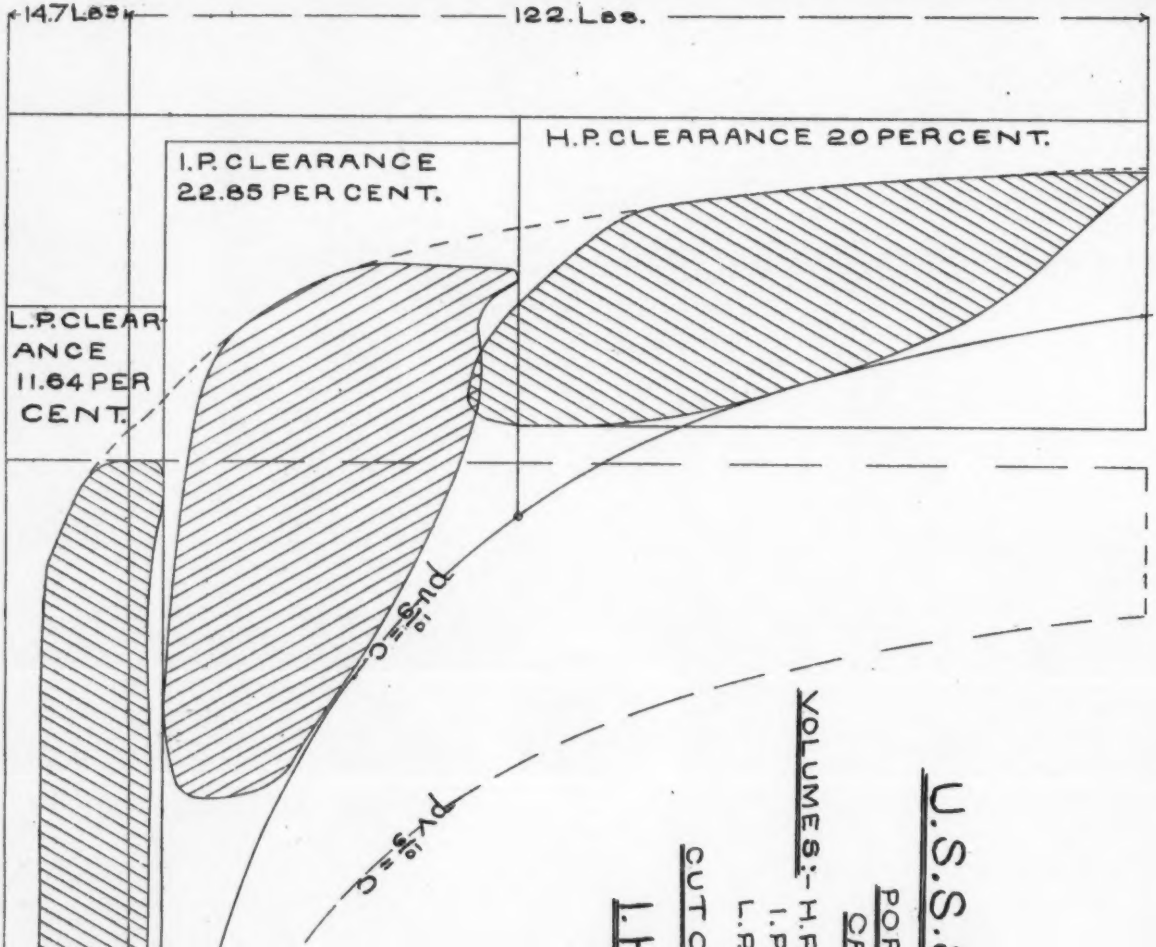


L.P. CYLINDER

STEAM PRESSURE 2ND RECEIVER 18.7 Lbs. ABS.



TOTAL I.H.P. = 4776.34


14.7 Lbs.
122. Lbs.
H.P. CLEARANCE 20 PER CENT.
I.P. CLEARANCE
22.85 PER CENT.
L.P. CLEARANCE
11.64 PER
CENT.
PV^1.0 = C
PV^1.3 = C
U.S.S.
POR
CA
VOLUME S:- H.P.
I.P.
L.P.
CUT O
I.H.

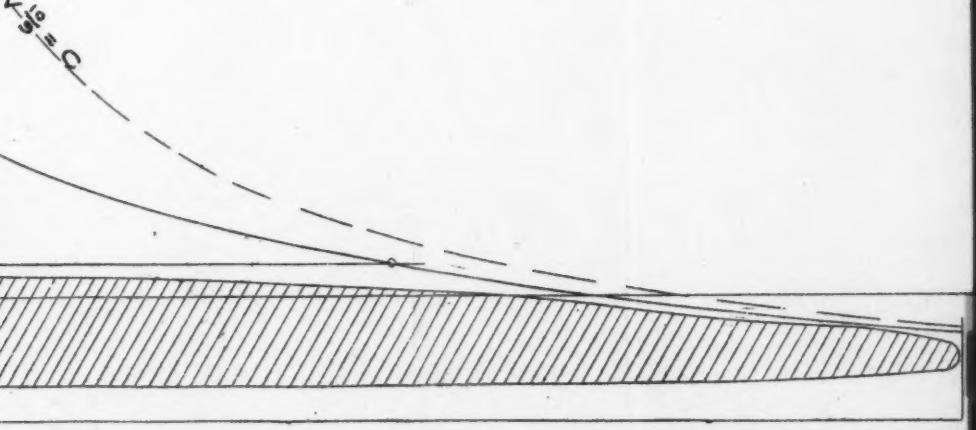
.S.S. SAN FRANCISCO.

PORT ENGINES
CARDS Nº 5.

ES:- H.P CYLINDER 28.52 CUBIC FEET.
I.P. " 58.6 " "
L.P " 144.53 " "

CUT OFF H.P. CYLINDER 0.69

I.H.P. 4776.34



locate the blowers so that they will draw the air from the whole fire room. In vessels with numerous bulkheads and a protective deck it is often practically impossible to secure this arrangement. I believe, however, that it is far superior to the closed fire room, and that it should be adopted wherever practicable.

My thanks are due to Assistant Engineer Frank W. Hibbs, U. S. Navy, for selecting and arranging the indicator cards, and constructing the combined diagrams.

TUBULOUS BOILERS.

(Continuation of Discussion.)

MR. MIERS CORYELL, ASSOCIATE.—In answer to Assistant Engineer F. C. Bieg, I insist that the weights I give for Belleville boilers of *Shearwater are the correct ones.* I know the Board's figures were obtained as correct, but they are not so.

I notice his understanding, as he states it, of the object of the trial, "to test the particular Belleville boiler of the yacht *Shearwater*, under certain conditions and limitations, * * * and to obtain some experimental figures of the only Belleville boiler in this country," and refer him to my previous comments upon the trial, its incompleteness and inadequate presentation of the boiler.

To the statements of Assistant Engineer S. H. Leonard, I have to say that, reasoning from the *Shearwater* trial, he does so from the standpoint of an incomplete and inadequate test trial, as I have stated.

The *Belleville* boilers in my sketch are made 14 feet high in a spirit of fairness, because I understood, from the engravings in the report of Com. Melville, that the space above the boilers was not available for coal stowage in that particular vessel; all the height needed for the Belleville boilers would be 11 feet 6 inches as compared with the 12 feet 8 inches for the Scotch boilers; see

engraving of Belleville boiler in No. 2, Vol. II, which is 10 feet 6 inches from fire-room floor to the highest part of the boiler, viz., the steam separator or drum. A glance will show abundant calorimeter above the tube elements and not project above this point. Therefore his diagrams of comparative spaces required are greatly in excess of what the Belleville boilers would occupy, and I also adhere to my fire-room space as sufficient from personal observation. Altogether, as to this, he must accept my statements.

As to the weights, he refers to the boilers of *Hirondelle*, *Milan* and *Voltigeur*. These groups are composed of generators having, say, 30 square feet of grate surface in each, not from a necessity of construction, but designed from a matter of opinion that with this area better firing can be done; using the 55 or 60 square feet furnace area, supposed to be desirable or permissible in the other types of tubulous boilers, and entirely practicable in the Belleville type, a very large reduction in weights would be obtained with result to approach nearly to the lighter types; always keeping in mind a better durability, the Belleville does not so greatly exceed the other types. There seems to be an impression that it is some fault of mine that this type of boiler has not been presented to the Government for a test trial. That is wrong. I have no pecuniary interest in the boiler. I have tried to have it investigated, unsuccessfully as yet, but I am prompted now to suggest that a good opportunity offers at the present time, as I read in the New York Times of the 4th inst., that trials of groups of 2,200 and 8,000 horse-power, are to be made soon by the French navy on vessels fitted with Belleville boilers.

The interest in the subject of tubulous boilers would seem to indicate that their use in groups of large power abroad extending, as it does, over 11 years, with increasing demand in both naval and merchant marine, it would be very desirable to obtain practical information through U. S. naval engineers of its working. Thus the requisite data would be acquired in the form prescribed by the Bureau of Steam Engineering, which is probably more complete in detail than that of any other navy.

[NOTE.—Mr. Coryell's reply to the criticisms on his discussion of the Belleville boiler, is, of course, cheerfully published, but attention must again be called to the fact that the weights of the *Shearwater's* boiler were obtained from the Atlantic Works of Boston, who put the boiler together and erected it in the vessel. Mr. Coryell's figures are only his estimate or calculation from the drawings or perhaps the estimate of the designers, and he can hardly expect his assertion to outweigh the scales. Also, as will be seen from the diagrams of Belleville boilers in groups (given elsewhere in this number), the fire rooms are all wider than shown in Mr. Coryell's sketches in the last number of the JOURNAL.—EDITOR.]

ASSISTANT ENGINEER J. H. CHALKER, U. S. R. M., ASSOCIATE.—I have read the papers published in the JOURNAL relating to tubulous boilers with much interest, especially those bearing upon the Ward and Belleville boilers.

When the Ward boiler referred to by Dr. Emery was first placed in the U. S. Revenue Steamer *Manhattan*, it was my fortune to be ordered to her, and I was on the vessel at the time when the bulge in the vertical column took place. In my opinion it was caused by improper circulation of the water, and aggravated by the lubricants used in the steam cylinders of the engine passing into the boiler with the feed water and being deposited upon the sides of the drum in proximity to the fire.

When the bulge was first discovered it was after a fast run, and the fires were forced a little more than usual to maintain the steam pressure. The water in the gauge during the run was not less than twenty inches, and when the engine was stopped fell to four inches, but this was not abnormal. Not only was the drum bulged, but all the seams in the lower section of the drum were leaking. When the bulge appeared in the new seamless sections, Assistant Engineer Whitworth, U. S. R. M., who was the engineer in charge, suggested that a hole be drilled in the bulge and a pipe fitted to it to lead out into the fire room, which was accordingly done.

When the engine was at rest and a valve at the end of the pipe opened, solid water flowed from it; but when the engine was

run at its maximum speed, steam mixed with water issued from the pipe, showing conclusively that the bulge was caused by overheating the section of the drum opposed to the hottest part of the fire, the overheating being due to poor circulation of the water in the drum.

A circulating pipe placed in the drum may prevent it from bulging more; but I have reason to believe that if the engine was again run at the same speed as before—that is, from 100 to 110 revolutions per minute—the result would be the same as in the first instance.

It would seem to me that the Ward type of boiler is not at all adapted for Government vessels, especially when used in vessels the size of our cruising cutters, on account of their inaccessibility for making repairs. If any derangement occurs to the vertical column it is no small matter to repair it, as was proved in the case of the *Manhattan*, as it necessitates the removal of the smoke pipe and boiler covering, breaking the joints on the manifolds, steam pipes, &c., and my judgment leads me to believe it would be the same with any of the lower coils on inside manifolds.

In regard to the accessibility for repairs, the Belleville boiler at least has the advantage of all other tubulous boilers designed for marine purposes that I have seen. This, in my opinion should eliminate any small advantage claimed for other types of boilers in regard to weight when used in men of war and revenue cutters.

MR. CHARLES WARD, ASSOCIATE.—Through the courtesy of the Secretary I have been favored with the proof of Mr. Chalker's remarks in regard to the trials of the Ward boiler on the revenue steamer *Manhattan*, and while I am satisfied that he intends to be perfectly fair, and that his opinion, though adverse to my boiler, is candid, I wish, with the permission of the Council, to place on record a few facts in order that the readers of the JOURNAL may be in possession of such information as will enable them to form an intelligent opinion in regard to this case.

I am very much interested in Mr. Chalker's graphic description of his experience with the Ward boiler on the U.S. Revenue

steamer *Manhattan*, and how he was ordered to her when the boiler was first put in her, and what his views are as to the cause of the bulging, etc. It is to be regretted that he did not give more detailed information as to the boiler problem; the duty developed by the *Manhattan*, both before and after the Ward boiler was introduced; how long the Ward boiler had been in use before this bulge developed, and how long the boiler has been in use since, so that engineers might form their own deductions. It would seem from his remarks that the Ward boiler was notorious only from the fact that it had bulged. Of course there is something wrong when a *bulge* is on hand. But the question is, What caused the bulge? We will weigh Mr. Chalker's views, and see what they are worth. In order to do this, it must be understood that the Ward boiler was under steam and accepted from the builder September 22, 1888. The *Manhattan* was put into commission upon anchorage duty, and did most excellent service until February, 1889.

Now the question naturally is, why the drum did not bulge until February, 1889, five months after the boiler had been in use? Is not the circulation in the boiler due to the conditions which existed when the boiler was first used? Or does it require five months to find out if the circulation in a boiler is perfect or not? Did it never appear to Mr. Chalker that the insidious grease, like the ardent spirit taken in drops, was gradually but surely, during these five months, increasing its hold upon the surfaces of the boiler until it had accumulated in sufficient quantity to prevent the water from contact with the metal, thereby leaving the metal subject to the action of the fire? There can be no doubt that the boiler worked perfectly until this. We have the testimony of the Chief Engineer, endorsed by the Captain and the Consulting Engineer, as follows, which speaks for itself:

OFFICE OF CONSULTING ENGINEER, U. S. R. M.,
22 CORTLANDT STREET,
New York City, January 8, 1889.

CHAS. WARD, ESQ.,
Charleston, W. Va.

Dear sir: I enclose copies of letters from Capt. D. F. Tozier

and Chief Engineer Stephen Keogh, of the U. S. Revenue Steamer *Manhattan*, certifying to the satisfactory working of the Ward boiler, recently applied to that vessel. I thought you would be gratified to know our opinion on this subject.

Very respectfully,

CHAS. E. EMERY,
Consulting Engineer, U. S. R. M.

U. S. REVENUE MARINE,
STEAMER *MANHATTAN*,
New York, January 8, 1889.

CHARLES E. EMERY, ESQ.,
Consulting Engineer, U. S. R. M.,
22 Cortlandt Street, N. Y.

My dear sir: I have the honor to herewith transmit for your information a letter addressed to me by Chief Engineer Stephen Keogh, U. S. R. M., concerning the efficiency of the new Ward boiler. I am much gratified to state that up to this time it has given the best possible results.

I am, sir, very respectfully, your ob't servant,

(Signed) D. F. TOZIER,
Captain, U. S. R. M.

U. S. REVENUE MARINE,
STEAMER *MANHATTAN*,
New York, January 8, 1889.

CAPTAIN D. F. TOZIER, *Commanding.*

Sir: In reply to your request of this date as to the efficiency of the new boiler, operating under the present trying conditions of alternately stopping and starting, I would respectfully state, in reply thereto, that the boiler works very satisfactorily. By using bituminous coal for fuel, the fires are easily managed and steam controlled. I find the boiler to be all that is claimed for it by its inventor, so far as it has been used.

Very respectfully, your ob't servant,

(Signed) STEPHEN KEOGH,
Chief Engineer, U. S. R. M.

It would seem from Mr. Chalker's statements that he must have been Assistant Engineer at this time; that being the case, it is surprising he should conclude the circulation was imperfect when the Chief Engineer stated voluntarily that "the boiler was all that is claimed for it," and the Captain endorsed the Chief

Engineer's statement by saying that "it had given the best possible results," all which the consulting engineer endorsed.

I quote this to show that for five months during the time Mr. Chalker was Assistant Engineer on the *Manhattan* the boiler must have had proper circulation or the above certificates would not have been given. Further, I would suggest if the bulging was due to imperfect circulation, why did it stop bulging? The metal must have been red hot to allow it to stretch. Such being the case, why did it not continue to stretch until it burst? If the circulation was imperfect when the plate was at its normal temperature—so imperfect that the plate was red hot—what prevented it getting hotter until it succumbed to the pressure?

Further, this drum in which the bulge developed is a vertical cylinder, 28 inches in diameter. Is it conceivable for a moment that a condition could exist whereby a circulation peculiar to itself would be established in a cylinder 28 inches diameter so as to allow the plates to heat if they were clean? Would not the very heat itself increase the circulation where there was nothing to impede the steam rising? Could anything be more simple than a cylinder of these dimensions? If there be this faulty circulation, as Mr. Chalker suggests, what would become of locomotive and marine fire-box boilers, where the water space is only from 1½ inches to 4 inches, at most.

Again, the central cylinder or drum in the Ward boiler is the heart, so to speak, of the whole boiler. All the water evaporated is fed into this cylinder; in addition to which, the water circulating through the boiler passes through the cylinder many times before it is evaporated.

The *Manhattan* engine is 19½ inches and 30 inches by 26 inches. With the former boiler the maximum duty was, according to my note book (dictated by Engineer Whitworth from the Engineer's log), 60 pounds of steam, cut off at half stroke. According to the report of the Board of Engineers Revenue Marine, of which Dr. Emery was president, the maximum performance of the engine with the Ward boiler was 115 pounds pressure, cut off at half and 101 revolutions, or within a fraction of double the duty developed with the former boiler. Under

these conditions there must pass through the central cylinder which bulged, no less than 8,508 pounds of feed water alone per hour. Inasmuch as the circulation is many times faster than the evaporation, it is highly probable that no less than 80,000 pounds passed through this drum each hour. Is it not at once evident that if only the amount of the feed passed through the drum (which is inevitable) it would amply remove the heat transmitted from about 21 square feet of heating surface?

It is quite obvious to a thoughtful mind that this trouble arises from the accumulation of grease; that the grease obstructed the transmission of the heat from the plate to the water, and consequently the plate heated and bulged, and would have continued bulging or stretching until it parted, did the evil not correct itself. For no sooner does the plate get hot than the grease disappears, at once admitting the contact of the water to the plate and checking the bulge until the grease again accumulates, when the operation is again repeated.

This condition of things was not, at first, apparent, but so soon as the internal surface was seen, the cause was evident. Potash was applied, the grease removed from the boiler, and no further trouble has occurred, although it is now ten months since the cleansing. Such being the facts, I am surprised that Mr. Chalker should at this late day renew the discussion, and endeavor to condemn the boiler for what would happen to every boiler with the same treatment, a condition which proper care, with our present experience, should never allow to occur. Dr. Emery, the Consulting Engineer of the U. S. Revenue Marine, in a recent paper published in this JOURNAL, (page 387,) states, in discussing this very case, "after cleaning the boiler thoroughly, *to remove the grease, no further difficulty* was experienced," showing that he agrees with the views taken by myself and almost every one who has investigated the matter.

As to the remark about drawing off *solid* water from the *active* part of any boiler under 100 pounds of steam, it hardly needs comment. Mr. Chalker's remarks on this matter, however, I think are inaccurate. They certainly do not agree with

the report of the Board before alluded to. The report says: "The water taken from this point" (*i. e.*, the pipe passing through the fire into the central drum, but cased in firebrick,) "was hotter and contained more steam than that taken from the bottom of the drum." Now the drum below the grates is for the deposit of mud, etc.; in it there is no circulation, except when blowing out. Advantage is taken of its location for the quiet which facilitates the deposit of foreign matter, and it would be very unnatural if the water at this point were not more solid. But who would expect to take solid water from a pipe tapped into the internal part of a fire box through three and a half feet of fire, even though it be encased in firebrick? Of course, on relief from pressure, much of the water would pass off as steam, which is the case even when trying a common gauge-cock.

I may say that Engineer Whitworth was of the same opinion as Mr. Chalker—that he could produce a bulge at any time by opening up the engine—and this test was made to show whether such was the case or not. I was present, and all I asked was that the grease should be removed. The engine was opened out; the fires were urged to all that natural draft and good firing with soft coal would do. With what result? I quote from the report of the Board: "Both the engine and boiler operated well, and there is no reason that this run should not be repeated at any time." I regret that Mr. Chalker was not present at this time, so that he might have seen for himself; for neither can he or anyone else produce a bulge on one of these boilers if the plates are clean.

The most conclusive proof of all, however, that the circulation in these boilers is perfect, and that the defect on the *Manhattan* was due to the grease deposit, is the test by the Coil Boiler Board under the direction of Chief Engineer Loring, U. S. N. In this case the boiler was under test for twenty-four hours, carrying from 160 pounds to 190 pounds of steam, burning 55 pounds of coal per square foot of grate, and evaporating about 8 pounds of water per square foot of heating surface. This is the severest test to which a boiler has ever been subjected, as Prof. Ken-

nedy's test of the Thornycroft boiler, and even the full power runs of torpedo boats are never over six or seven hours' duration. Moreover, before the actual test, the Ward boiler was under steam several days with forced draft. Notwithstanding this exceedingly severe test—compared with which that on the *Manhattan* was hardly more than banked fires—there was no evidence of the slightest stress on any part of the boiler. How severe this test was may be shown by the fact that the evaporation of the Ward boiler *was nearly double that of the recent test on the Thornycroft* boiler on the *Cushing*, as reported in the New York Times and other papers, although both boilers have practically the same heating surface.

I do not claim that the Ward boiler is perfect. Such a boiler has not yet been, and probably never will be designed. Nor do I claim that my boiler will defy the laws of nature, which is what was apparently expected in the case of the *Manhattan*. It is the commonest thing in the world for a crown sheet to bulge and "come down" when a deposit of grease has been allowed to accumulate, although no one has yet appeared to attempt an explanation of such cases on the ground of defective circulation.

BELLEVILLE BOILER.—In accordance with the statement on page 405 of the last number of the JOURNAL, the following additional information is given from data furnished by Mr. Belleville:

Test of a land boiler at Nice, France, May 9, 1884.—Grate surface, 39.72 square feet; heating surface, 1,264.2 square feet; ratio of heating to grate surface, 31.83. Fuel, briquettes du Sud-Est, (made at Port-de-Bouc,) and from experiment found to have about 92 per cent. of the thermal value of good Cardiff coal.

Duration of test	4 hrs. 5 mins.
Steam pressure in boiler	156.3 lbs.
Steam pressure after passing reducing valve	93.1 lbs.
Draft (in tube box) in inches of water	$\frac{1}{8}$
Total coal burned	2,211.2 lbs.
Total refuse in ashes, clinker, &c	227.9 lbs.
Total combustible consumed	1,983.3 lbs.
Coal per hour per square foot of grate	12.23 lbs.
Temperature of feed water (Fahr.)	63.5°
Temperature of steam (Fahr.)	361.4°

Total amount of water vaporized		20,486 lbs.
From temperature of feed at temperature of steam.	Vaporization per hour	5,020 lbs.
	Vaporization per hour per square foot of G. S.	126.4 lbs.
	Vaporization per hour per square foot of H. S.	3.97 lbs.
	Vaporization per hour per pound of coal	9.62 lbs.
From and at 212° Fahr.	Vaporization per hour	5,757 lbs.
	Vaporization per hour per square foot of G. S.	144.95 lbs.
	Vaporization per hour per square foot of H. S.	4.55 lbs.
	Vaporization per hour per pound of coal	11.03 lbs.

[These results, while showing the boiler to be economical, are not extraordinary, as an examination of the tests of a number of Babcock and Wilcox boilers have shown results somewhat higher.]

It should be stated that this test was made by "judging" the condition of fire, water-level, &c., at beginning and end of test, and the results are consequently somewhat higher for total evaporation than when beginning and ending with bare grates.

In connection with the foregoing test, a chemical analysis of the feed water before and after its passage through the "purifier" was made, with the following result, the figures being in grammes per litre of the water:

	Before admission.	After exit.	Per cent. of salt precipitated.
Carbonate of lime	0.06	0.01	83
Sulphate of lime	0.336	0.014	96
Magnesia	0.012	0.012	0

TESTS OF MACHINERY OF THE CRUISER *RIGAULT DE GENOUILLY* UNDER NATURAL DRAFT.

Total grate surface	252 square feet.
Total heating surface	7,416 square feet.

	Half-power, June 18, 1889.	Full power, June 22, 1889.
Duration of test	6 hrs.	6 hrs. 6 min.
I. H. P. developed	1,272	2,280
Coal per hour per I. H. P.	2.37	2.04
Evaporation per hour per pound of coal	8.3	9.47
Coal per hour per square foot of grate	23.95	18.51

In the half-power trial the boilers were pushed to their utmost under natural draft. In the full-power trial the firing was about that expected in good average conditions.

M. Belleville has kindly furnished the particulars of his boilers, as actually fitted or designed for some of the more recent French war vessels, from which the accompanying table has been compiled.

It will be observed that nothing is said of the capabilities of the boilers under forced draft. This is because data of the intended forced draft I. H. P. were obtainable for only one set of designs. M. Belleville says on this point, in a letter to the Secretary, "We consider working under an energetic artificial draft an unfavorable condition for any kind of boiler. The precise advantage in the use of our boilers is to permit the placing in a given space of a greater area of grate surface than in cylindrical boilers, and consequently to secure the same power with a smaller amount of coal burned per square foot of grate."

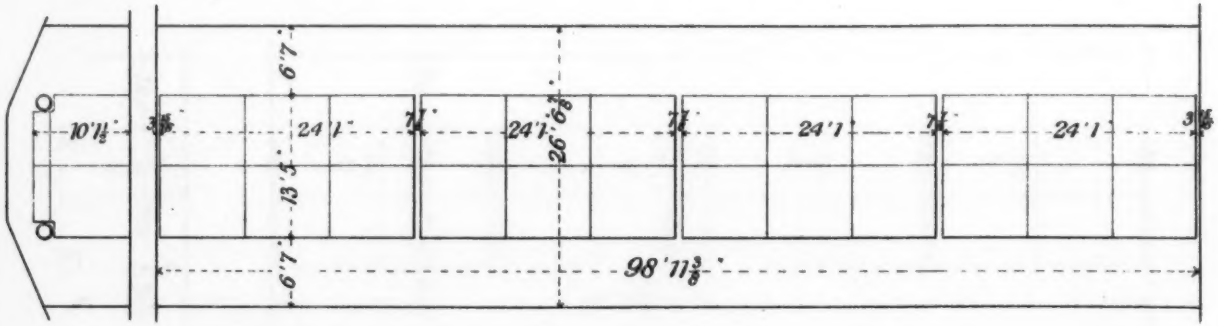
It should be added that a great many Belleville boilers have been placed in the naval vessels of France and other countries for auxiliary purposes.

The accompanying diagrams of the arrangement of boilers for French war vessels were furnished by Mr. Belleville.

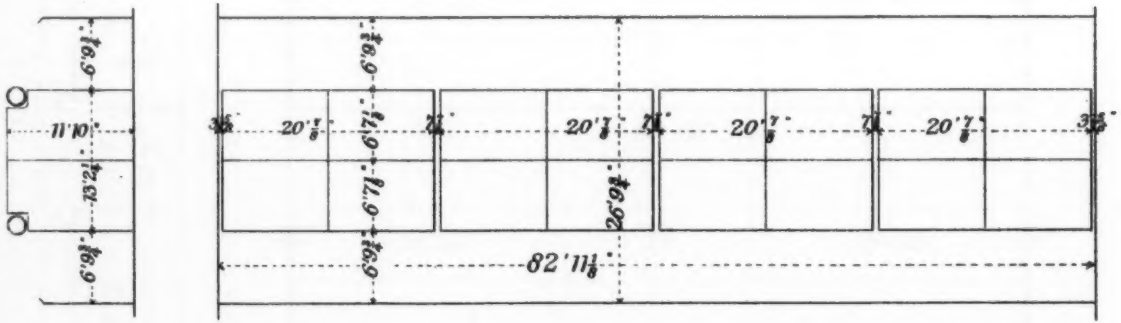
Name.	I. H. P. natural draft.	G. S. square feet.	H. S. square feet.	Width between coal bunkers.	Length of fire room between bulkheads.	Weight including water and brickwork, but excluding chimney.	Weight per I. H. P. natural draft.	Weight per square foot H. S.
				feet.	*feet.*	*tons.*	*I. H. P. per ton.*	*pounds.*
Milan	3,945	365.1	11,130	25.75	49.08	155.52	25.37	31.30
Hirondelle	2,071	239.8	7,420	25.75	33.86	111.72	18.54	33.72
Rigault de Genouilly		252.1			32.48	120.08	17.25	36.25
Alger	7,890	753.5	22,265	26.58	98.95	342.55	23.04	34.46
Latouche-Treville	7,300	694.4	19,525	26.97	82.94	303.30	24.07	34.80
Chanzy								
Charner								
Leger	2,170	209.9	5,630	14.51	51.60	80.32	27.02	31.95
Levrier								
Brennus	13,810	1,100.5	35,030	49.05	69.68	529.6	26.08	33.87

The I. H. P. is in English measure, which accounts for the lack of round numbers.

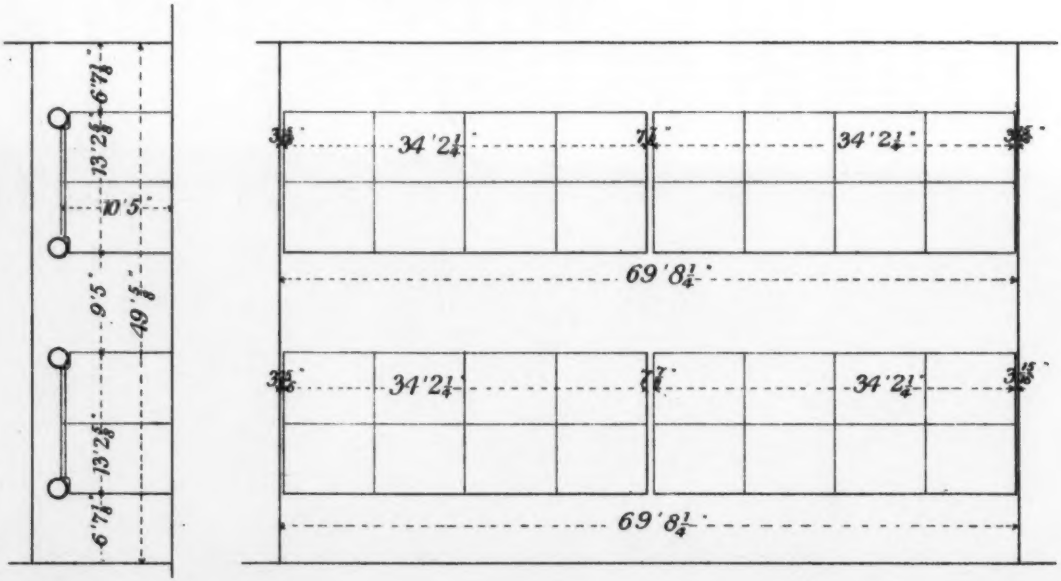
Arrangement of Boilers on the Belleville System
for the Cruiser "Alger,"
of the French Navy.



Arrangement of Belleville Boilers for the
Armored Cruisers 'Chanzy and Latouche Tréville,'
of the French Navy.



Arrangement of Boilers on the Belleville System
for the Battle Ship "Brennus",
of the French Navy.



NOTES.

MARTIN'S SYSTEM OF INDUCED DRAFT.

Martin's arrangement for induced draft consists of one or more fans placed in the base of the smoke pipe and driven by means of friction gear at a high number of revolutions. By means of diaphragms and dampers suitably arranged, the escaping gases may be directed either directly up the smoke pipe, when natural draft is used, wholly through the fans when the latter are in full operation, or in part through the fans and in part up the pipe.

A description of this system, which is practically identical with the arrangement adopted by Kemp, of the firm of Stephen & Sons, in Glasgow, in connection with his feed-water heater, and introduced by him in 1886 on the French freight steamer *Bléville*, was given at the meeting of the Institute of Naval Architects in 1887, by McFarlane Gray.

The advantages then claimed were:

1. The possibility of doing away with the smoke pipe on board war vessels, because of having the escape gases under control to direct them where convenient.

2. Simplicity as compared with other systems for accelerating the draft.

3. Reduced wear of the fire tubes.

This last point is the chief merit the system has, and finds its reason in the fact that the flames do not impinge on the tube sheet and the tube ends, but split into tongues before reaching the mouths of the tubes.

With these advantages McFarlane Gray contrasts these disadvantages:

1. Owing to increased temperature and volume of escape gases, the fans must be larger than those used with forced draft systems.

2. The uptakes must be incrased in bulk and weight to accommodate the fans.

Test of Martin's System of Induced Draft.

Date of trial.	Duration of trial.	Description of coal.	Average pressure of steam in lbs. per sq. in.	Unbalanced air pressure in inches of water.			Temperature in Fahr. degrees.		Revolutions per minute.		Cubic feet of air used per sq. ft. of grate per min.	Pounds of water evaporated per pound of coal.		Pounds of coal consumed per sq. ft. of grate per hour.	Pounds of water evaporated per square foot of grate per hour.		I. H. P.
				Smoke box.	Eye of fan.	Base of funnel.	Feed.	Atmosphere.	Engine.	Fan.		At feed temperature.	From and at 212° Fahr.		At feed temperature.	From and at 212° Fahr.	
July 25, 1890.....	8 hrs.	Harris's Nav.	74.3	0.57	0.85	.25	60.9	68.8	84.6	376.4	335.0	10.02	11.58	30.69	307.0	355.0	336.3
July 26, 1890.....	8 "	" "	74.68	1.25	1.43	.258	60.4	67.9	101.3	445.0	340.0	9.3	10.75	42.7	397.0	459.0	434.3
July 28, 1890.....	8 "	" "	76.56	1.55	1.87	.57	60.87	68.1	124.9	534.7	345.0	9.09	10.51	49.95	454.0	525.0	496.8
Aug. 1, 1890.....	8 "	" "	74.3	2.06	2.5	.54	62.0	67.3	139.8	636.9	352.4	8.3	9.58	63.7	528.0	610.0	579.0
Aug. 5-9, 1890.....	96 "	" "	74.2	1.26	1.56	.33	62.1	69.9	104.0	485.0	343.4	9.64	11.13	40.4	389.6	450.37	426.0

Ashes, 4.42 per cent.; clinkers, 1.47 per cent.

3. The fans are apt to give trouble working in temperatures as high as 900° Fahr.

Martin claimed at the time that these disadvantages did not exist in the extent noted by Mr. Gray, and instanced the steamer *Olive Branch*, which has been in successful operation for eighteen months. The fan is 6 feet in diameter, does not necessitate an extraordinary enlargement of the uptake, and has given no trouble working in a temperature of 450° Fahr. Martin claims that a fan making four hundred or more revolutions protects itself from the heat. Busley makes the prediction that Martin's system will, in all probability, find its application, on smaller boiler plants and with temperatures not excessive, *i. e.*, with comparatively light draft.

During the month of August a series of trials was made under the auspices of the British Admiralty to determine the merits of the Martin system. One of the original boilers of the *Polyphemus*, of the locomotive pattern, was chosen for the trial. The boiler has two furnaces, separated by a water partition. Total effective grate surface, 20.77 square feet. Heating surface, about 758 square feet. The fan used was 4 feet 1 inch in diameter and 1 foot 6 inches wide. The system was tested for eight consecutive hours on each of four days, and finally for ninety-six hours, from August 5th to August 9th, with intermissions for cleaning fires only. There appears to have been no trouble from depositon of carbonaceous matter on the fan blades, nor from the heat of the gases. The results are given in the table on opposite page.

COMPARATIVE TRIALS OF BOILERS FITTED WITH ORDINARY AND WITH THE "SERVE" BOILER TUBES.

The "Serve" tube differs from the ordinary tube in having inwardly-projecting ribs from $\frac{3}{8}$ to $\frac{1}{8}$ inch long, according to the size of the tube, nearly doubling the heating surface in contact with the gases. Until recently they were made of copper or brass only, but they are now successfully manufactured of steel. Plates are rolled of the required length and breadth having eight projecting ribs. These plates are then bent into tubes and lap welded.

In England a steamer, the *Phra Nang*, built by the Fairfield Shipbuilding and Engineering Company, has just been fitted out with "Serve" tubes, and will soon complete an outward voyage to Hong Kong. As a sister ship, fitted with ordinary tubes, started three weeks earlier, the results as to speed and coal consumption will be of interest as bringing out a point left unsettled by the tests, namely, to what extent will the increased thermal resistance due to formation of deposits on the water side of the tubes modify the test results. In France the Compagnie Générale de Navigation adopted the tube for their steamer *Bourdon* five years ago, and have since fitted out eight more with it. The French Admiralty are testing the tube in the boilers of a cruising ship. The Compagnie Générale Transatlantique are also adopting the novelty. It has besides been tried on locomotives, and in general has been found to result in economy wherever accelerated draft was used.

The trials, the results of which are given in the tables following, were made at Sheffield, under the supervision of representative engineers. Trials were made both with natural draft and with induced draft on Martin's system. Satisfactory arrangements were made for measuring the water and weighing the coal, and evaporation took place with the safety valves removed, or under atmospheric pressure increased by that due to the flow of steam through the contracted openings (two of $3\frac{1}{2}$ inches diameter). The air pressure in the uptakes and the velocity of air to the ash pits were recorded at least every half hour. The revolutions of the fan do not appear in the tabulated results, for the reason that they were not taken direct but calculated from those of the engines, which drive the fans by friction gearing.

Nixon's navigation coal was used in all the trials.

Of two boilers, of identical design, No. 1 was fitted with the "Serve," No. 2 with ordinary tubes. The following table gives the principal dimensions of the boilers:

	No. 1.	*No. 2.*
Diameter	10 feet 6 inches.	10 feet 6 inches.
Length	10 feet 6 inches.	10 feet 6 inches.
Inside diameter of furnaces (2 in each boiler)	2 feet $10\frac{1}{2}$ inches.	2 feet $10\frac{1}{4}$ inches.

	No. 1.	*No. 2.*
Total number of tubes	126.	126.
Outside diameter of all tubes	3½ inches.	3½ inches.
Thickness of ordinary tubes	⅛ inch.	⅛ inch.
Number of stay tubes	34.	34.
Thickness of stay tubes	⅛ inch.	⅛ inch.
Length of furnaces and of all tubes inside of boiler	7 feet 6½ inches.	7 feet 6½ inches.
Heating surface of each ordinary tube	10.42 square feet.	5.95 square feet.
Heating surface of each stay tube	10.42 square feet.	5.45 square feet.
Total heating surface of tubes	1,312.9 square feet.	732.7 square feet.
Area of grate	31 square feet.	31 square feet.
Heating surface of furnaces	135.5 square feet.	135.5 square feet.
Heating surface of combustion chamber	88 square feet.	88 square feet.
Total heating surface of boilers	1,536.4 square feet.	956.2 square feet.
Proportion of grate to heating surface	1 to 49.56	1 to 30.84
Calorimeter of tubes	802.36 square inches.	852.18 square inches.
Section of fire bars	1 inch square.	1 inch square.
Distance between fire bars	½ inch.	½ inch.
Smoke pipe, rectangular	4 feet × 2 feet.	4 feet × 2 feet.
Height of smoke pipe from top of combustion chamber	43 feet 7 inches.	43 feet 7 inches.

From this table it is apparent that while the "Serve" tube-boiler has the advantage of greater heating surface, it has a decreased calorimeter. It is the heavier boiler of the two, but is lighter for equal area of heating surface.

TRIAL NO. 1.—DIFFERENTIAL PRESSURE OF ONE-HALF INCH IN EACH SMOKE BOX.—DURATION, THREE HOURS.

	No. 1.	*No. 2.*
Total coal burnt	34 cwt.	37 cwt.
Evaporation in ½ hour	345 gallons.	310 gallons.
" 1 hour	935 "	870 "
" 1½ hours	1,555 "	1,465 "
" 2 hours	2,115 "	2,030 "
" 2½ hours	2,645 "	2,510 "
" 3 hours	3,250 "	3,100 "

Boiler No. 1 gave 4.8 per cent. greater evaporation with less coal, the economy in coal, taking into account the increased evaporation, being 14.1 per cent.

TRIAL NO. 2.—NATURAL DRAFT.—DURATION, THREE HOURS.

	No. 1.	*No. 2.*
Total coal burnt	16 cwt. 18 lbs.	20 cwt.
Evaporation in 1 hour	605 gallons.	640 gallons.
" 1½ hours	920 "	980 "
" 2 hours	1,210 "	1,300 "
" 2½ hours	1,510 "	1,680 "
" 3 hours	1,820 "	2,010 "

In this instance the evaporation of No. 1 boiler falls short of the other by 9.4 per cent.; but the coal economy of No. 1 over No. 2 was 12½ per cent. Both these trials were too short to furnish absolute results, but were fair by way of comparison. There was no more sooty deposit in the "Serve" than in the ordinary tubes.

TRIAL NO. 3.—INDUCED DRAFT REGULATED TO EQUALIZE COAL CONSUMPTION IN THE TWO BOILERS.—DURATION, TWELVE HOURS.

Boiler No. 1.

Hours.	Coal burnt in pounds.	Water evaporated in pounds.	Vacuum at base of chimney	Steam pressure in pounds.	Temperature of feed.
1	1,568	9,000	1¼ inches.	16½	59°
2	2,800	19,300	1 inch.	12½	66°
3	3,808	27,000	1¼ inches.	15	60°
4	4,928	39,800	" "	11	61°
5	5,824	49,600	1 inch.	10	60°
6	6,832	59,300	1¼ inches.	11	60°
7	7,504	68,600	" "	10	60°
8	8,400	77,200	" "	10	60°
9	9,520	87,300	" "	11¼	60°
10	10,416	96,000	" "	10½	60°
11	11,200	105,800	1⅛ "	10¼	60°
12	11,872	114,600	1 inch.	10	60°

Boiler No. 2.

Hours.	Coal burnt in pounds.	Water evaporated in pounds.	Vacuum at base of chimney.	Steam pressure in pounds.	Temperature of feed.
1	1,568	7,800	1¼ inches.	9½	59°
2	2,800	16,200	" "	8½	61°
3	3,808	25,300	1½ "	9	61°
4	4,928	34,400	1½ "	6	62°
5	5,824	43,000	1½ "	6	61°
6	6,832	52,300	1½ "	10	60°
7	7,504	61,500	1⅛ "	6½	61°
8	8,400	69,300	" "	6¼	60°
9	9,520	78,500	1¼ "	8	60°
10	10,416	87,500	1 inch.	6¼	60°
11	11,200	95,100	1⅛ inches.	4½	61°
12	11,872	103,000	" "	6¼	60°

The "Serve" tube boiler evaporated 11,600 pounds, or 11.26 per cent. more than No. 2. The evaporation from and at 212° Fahr. was 11.19 pounds for No. 1 and 10.05 pounds for No. 2.

TRIAL NO. 4.—DIFFERENTIAL PRESSURE MAINTAINED, ¾ INCH.—COAL CONSUMED, THE SAME.—DURATION OF TRIAL, SEVEN HOURS.

Boiler No. 1.

Time.	Water evaporated, in lbs.	Vacuum in eye of funnel.	Steam pressure.	Temperature of feed water.	Velocity of air in ash-pits in feet per minute.	
					Right furnace.	Left furnace.
10.5—start.						
10.30	2,800	⅛ in.	12	61°	985	940
11.0	7,500	⅜ in.	11	63°	850	900
11.30	12,900	"	12	61°	795	805
12.0	17,600	"	12¼	60°	605	740
12.30	22,800	"	9¾	64°	605	625
1.0	26,900	"	9	64°	695	830
1.30	31,600	"	9¼	63°	695	800
2.0	35,400	"	9	62°	480	650
2.30	39,800	"	9	63°	520	760
3.0	44,300	"	8¼	63°	490	580
3.30	48,800	"	8¼	62°	520	655
4.0	53,000	"	9	63°	570	650
4 30	56,600	"	9¼	62°	560	630
5.5	61,500	"	9	62°	570	560

Boiler No. 2.

Time.	Water evaporated. in lbs.	Vacuum in eye of funnel.	Steam pressure.	Temperature of feed water.	Velocity of air in the ash-pits in feet per minute.	
					Right furnace.	Left furnace.
10.5—start.						
10.30	2,350	⅜ in.	6¼	62°	780	860
11.0	6,300	⅜ in.	8¼	64°	905	800
11.30	11,400	"	6½	66°	680	820
12.0	15,300	"	6¾	66°	600	620
12.30	20,100	"	4½	63°	540	580
1.0	23,400	"	4¼	62°	535	680
1.30	27,400	"	4½	63°	540	500
2.0	30,400	"	4¼	62°	490	610
2.30	34,300	"	4½	62°	435	690
3.0	38,200	"	4	63°	490	585
3.30	42,100	"	4¼	64°	480	580
4.0	45,800	"	4½	63°	460	680
4.30	48,800	"	5	62°	460	565
5.5	53,200	"	4¼	62°	400	550

No. 1 showed an excess in evaporation over No. 2 of 15.6 per cent. Evaporation from and at 212°, for No. 1 boiler, 11.03 pounds; for No. 2 boiler, 9.5 pounds per pound of fuel, an excess of 16.1 per cent. in favor of No. 1. The smoke-box temperatures of No. 1 were uniformly lower than those of No. 2. About

the same quantity of soot was deposited in the lower row of tubes.

TRIAL NO. 4.—DIFFERENTIAL PRESSURE MAINTAINED IN NO. 1, 5/8 IN.; IN NO. 2, 1/2 IN. THIS MORE THAN COMPENSATED FOR THE DIFFERENCE IN CALORIMETER.—DURATION, SEVEN AND ONE-QUARTER HOURS.

Boiler No. 1.

Time.	Water evaporated, in lbs.	Vacuum in eye of funnel.	Steam pressure, in lbs.	Temperature of feed water.
1st hour.	6,600	5/8 in.	9	65°
2d "	15,200	"	8¼	62°
3d "	23,300	"	7	66°
4th "	31,300	"	7½	64°
5th "	39,100	"	7	64°
6th "	45,300	"	6¼	63°
7¼ hours.	56,100	"	6¾	63°

Coal consumed, 5,712 lbs.

Boiler No. 2.

Time.	Water evaporated in pounds.	Vacuum in eye of funnel.	Steam pressure in pounds.	Temperature of feed water.
1st hour.	6,000	½ inch.	4½	68°
2d "	13,400	"	4¼	65°
3d "	20,000	"	3¾	64°
4th "	27,100	"	4	65°
5th "	33,500	"	3	64°
6th "	38,600	"	3¼	63°
7¼ hours.	48,400	"	3	63°

Coal consumed, 5,404 pounds.

Burning 5.7 per cent. more coal, the "Serve" tube boiler evaporated 15.86 per cent. more water than the boiler with ordinary tubes. Evaporation from and at 212° Fahrenheit, for No. 1, 11.39 pounds; for No. 2, 10.36 pounds per pound of fuel, a difference in favor of No. 1 of 9.94 per cent.

SHIPS.

UNITED STATES.

Battle Ships.—(Described on page 409 of current volume.) Bids for the construction of these vessels were opened October 1. The proposals were as follows:

The Wm. Cramp & Sons' Ship and Engine Building Co., Department's designs, one ship, $2,990,000; two ships, $2,890,000 each. Contractors' hull, retaining Department's machinery, one ship, $3,120,000; two ships, $3,020,000 each.

The Union Iron Works, of San Francisco, Department's designs, one ship, $3,240,000; two ships, $3,200,000 each.

The Risdon Iron Works, of San Francisco, Department's designs, one ship, $3,275,000.

The Bath Iron Works, of Bath, Maine, Department's designs, one ship, $3,149,000.

Accompanying the bid for their own design, the Messrs. Cramp submitted a statement showing in a general way the changes they would make if authorized to build on those designs. The ship was to be lengthened 12 feet, thus increasing the normal displacement 480 tons. This increase was to be distributed thus: 190 tons for increase of hull due to added length, the remainder to be used at the discretion of the Department in heavier side armor or heavier around the gun stations.

From the terms of the appropriation one of the ships must be built on the Pacific Coast, so the awards were made—two ships to the Messrs. Cramp and one to the Union Iron Works.

Since the opening of the bids it has been decided to enlarge these ships even more than the proposition of the Messrs. Cramp.

As already stated, the machinery designs remain unchanged. The dimensions of the hull are now as follows: Length on load water line, 348 feet; extreme beam, 69 feet 3 inches; mean draught, 24 feet; displacement, 10,200 tons. The expected maximum speed is 16.2 knots, and the sustained sea speed 15 knots.

The battery will be increased by the addition of two 8-inch rifles and two 1-pounder rapid-fire guns.

The thickness of armor is as follows: Main belt, 18 inches; end diagonals, 14 inches; casemates, 5 inches; 6-inch B. L. R., local protection, 5 inches; 13-inch guns, redoubts and turrets, 17 inches; 8-inch guns, barbettes, 8 and 10 inches; turrets, 6 and 8½ inches; conning tower, 10 inches; armored deck, 2¾ and 3 inches.

The price for these enlarged vessels was fixed on the basis of the bid of the Messrs. Cramp at $3,020,000 for each of the two awarded to them. The price of that to be built by the Union Iron Works was fixed at $3,180,000, the difference between the two amounts being on account of the additional cost of the material due to the freight charge for transportation from the East to San Francisco.

Protected Cruiser No. 12 (the Triple-Screw Ship).—This was described on p. 409 of current volume. The bids were opened at the same time as those for the battle ships, and were both for the Department's designs throughout. The appropriation for this vessel is $2,750,000.

The proposals were:

The Wm. Cramp & Sons' Ship and Engine Building Co., $2,725,000; the Union Iron Works, $3,025,000.

The bid of the Messrs. Cramp being the lowest, and the only one inside the appropriation, the contract was awarded to them.

Harbor Defense Ram.—This vessel, commonly known as the *Ammen Ram*, has been designed by the Navy Department in accordance with the plans and suggestions of Admiral Ammen, under the provisions of an Act of Congress of March 2, 1889. She will be similar in many respects to the *Polyphemus* of the British navy, though not quite so large. The bow will be shaped like a parabolic spindle, to give strength for ramming, and the cross-sections are approximate ovals, the draught of water being such that there will be very little freeboard. The rounded upper surface will offer a very poor target.

The hull dimensions are: Length over all, 243 feet; length on load water-line, 242 feet 9 inches; extreme beam, 43 feet 5

inches; beam at water-line, 41 feet 10 inches; mean draught, 15 feet; displacement, 2,050 tons. The expected speed is 18 knots; contract speed, 17 knots. There is no armament.

The machinery consists of horizontal, triple-expansion engines driving twin screws, and placed in separate water-tight compartments divided by an athwartship bulkhead. The cylinders are 25, 36 and 56 inches diameter, by 36 inches stroke. The I. H. P. of the main engines, and the air and circulating pump engines, is expected to be about 4,800 when the main engines are making 150 revolutions per minute. The valves of the main engines will all be of the piston type driven by Marshall valve-gear. There will be one for each H. P. and I. P. cylinder, and two for each L. P. cylinder.

The condensers will be of composition and sheet brass, with a cooling surface of 3,334 square feet each. For each condenser there will be two vertical, single-acting air pumps and a horizontal, double-acting circulating pumps, all worked by a horizontal steam cylinder.

There will be four boilers of the low cylindrical (marine locomotive) type, 11 feet 6 inches in diameter and 19 feet 3 inches long, constructed for a working pressure of 160 pounds. They will be in two separate water-tight compartments. Each boiler will have three corrugated furnaces, 3 feet 8 inches in internal diameter. The total grate surface will be 288 square feet, and the total heating surface 10,797 square feet. The forced draft will be by closed fire rooms.

Special provision will be made by chocks and stays to prevent the machinery and boilers shifting when the vessel is ramming.

The proposals for this vessel and the *Torpedo Boat* (No. 2) will be opened December 20, and those for *Torpedo Cruiser No. 1*, February 11, 1891.

Maine.—This vessel was successfully launched at the New York Navy Yard on November 18th. This marks a distinct event, as the *Maine* is the first steel vessel ever launched from one of our Navy Yards, and she is, further, the first modern battle ship of the United States Navy to take the water. As shown in our last number, she is still far from finished, and it

will probably be at least two years before she will be tried and commissioned.

Concord.—An official trial of this vessel took place November 20, but was unsuccessful. The course was in Long Island Sound, and the weather was entirely favorable. For the first hour everything worked well, the engines making at times 156 revolutions, which, as the pitch of the screw is slightly greater than that of the *Yorktown*, indicates that she was developing a greater I. H. P. than that vessel's average. As the trial proceeded, however, the steam pressure fell, and with it the revolutions. It is stated that this was due to trouble with the feed pump for the forward boiler, which would not function properly if the pressure was higher than 140 pounds. The pressure was therefore allowed to drop to about this amount, at which it was maintained until the last hour, when water began to come out of the ash-pits of the starboard after boiler and to fill up the air duct. This necessitated shutting off the forced draft from this boiler, the steam dropping to 110 pounds.

The main engines and air and circulating pumps worked perfectly and without any indications of heating even at the highest speed. This good working of the air pumps is worthy of note because there has been a good deal of trouble on some of our recent ships with these pumps of the crank and fly-wheel type. The air pumps on the *Concord* are driven through a beam by a simple engine with the direct-acting-pump valve-gear.

For the first hour and a half the speed, as determined by bearings on shore and against a tide estimated at one and a half knots, was 17.6 knots per hour through the water.

It is a matter of great regret that the trial was not a success, especially as it was the first time in the forced draft trial of any vessel that the coal was accurately measured *during the full-power trial.* The contractors for the *Concord*, with a zeal for securing exact engineering data which deserves the thanks of the profession, arranged for an accurate record of the coal consumption by putting up about 30 tons in bags of 100 pounds each. A careful tally was kept of these bags as emptied, and after the trial they were again counted. The value of correct in-

formation on this point cannot be overestimated, as hitherto we have had to depend entirely on estimates which have not been at all in agreement for different ships. When the next trial occurs, which will in all probability be a complete success, the data of coal per I. H. P. per hour will be received with great interest.

The following are some of the data of the trial, and it should be noted that the *Concord's* machinery is a duplicate of the *Yorktown's*, except that she has closed ash-pit draft, while the latter has closed fire rooms.

The trial began at 9.30 a. m. For the fifteen minutes beginning at 9.45 the averages were: Starboard engine—revolutions, 150.9; steam at boilers, 163.5; at engines, 158; at first receiver (abs.), 80; in second receiver (abs.), 30; cut off at three-fourths; vacuum, 24 inches. Port engine—revolutions, 153.9; steam at boilers, 163.5; at engine, 158; in first receiver (abs.), 81; in second receiver (abs.), 31; cut off at three-fourths; vacuum, 24 inches. The I. H. P. (without correction) of the main engines was 1,700 starboard and 1,650 port, and 200 (estimated) for the auxiliaries, or a total of 3,550.

For the fifteen minutes beginning at 12.15 p. m.: Starboard engine—revolutions, 141.8; steam at boiler, 142; at engine, 135; in first receiver (abs.), 71; in second receiver (abs.), 27.5; cut off at three-fourths; vacuum, 24 inches. Port engine—revolutions, 144.8; steam at boilers, 142; at engine, 138; in first receiver (abs.), 73; in second receiver (abs.), 28; cut off at three-fourths; vacuum, 24 inches. The I. H. P. (without correction) of main engines was 1,320 starboard and 1,360 port, and the auxiliaries (estimated) 200, or a total of 2,880.

For the fifteen minutes beginning at 1.15 p. m.: Starboard engine—revolutions, 130.3; steam at boilers, 111.5; at engine, 106; in first receiver (abs.), 57; in second receiver (abs.), 23; cut off at three-fourths; vacuum, 24 inches. Port engine—revolutions, 133.1; steam at boilers, 111.5; at engines, 108; in first receiver (abs.), 58; in second receiver (abs.), 24; cut off at three-fourths; vacuum, 24 inches.

During the trial the temperatures were: feed, 124° to 130°; discharge, 107°; injection, 54°.

The pitch of the *Concord's* screw is 13 feet 2½ inches; that of the *Yorktown* on her trial was 12 feet 6 inches. (For account of this trial see pp. 137–146 of vol. I.)

Cushing.—A very interesting series of trials of one of the Thornycroft boilers of this vessel has recently been conducted with a view to obtaining accurate information on the potential and economic evaporative power at various rates of combustion.

Four trials were made, two of eleven and a half hours each, with an air pressure of one-half inch and three inches, respectively, to ascertain the capacity of the boiler for prolonged steaming; another with the highest possible air pressure and most rapid firing, to determine the greatest possible power for a "spurt" of about an hour; and, finally, a trial of about two hours and a half, with the blowers discharging into open fire rooms, to find the most economical rate of combustion.

The experiments were made with great care under the general direction of Chief Engineer Charles H. Loring, U. S. Navy, who also directed those of the Ward and Cowles boilers, of which data are given on pp. 414–416 of current volume. Every care was taken to insure accuracy. The feed water was measured by a Worthington water meter, kindly loaned by the Worthington Pump Company, which was carefully standardized by tests at different speeds, and found to have a very slight and constant error. It may be remarked, in passing, that the use of a reliable meter has decided advantages over tank measurements, in that, its record being continuous, readings can be taken at any time, and the data of feed water thus recorded every fifteen minutes or half hour, when the coal and other data are taken. The coal was weighed into bags, and at least four counts by different observers obtained, thus insuring accuracy. The ashes were also carefully weighed. Calorimetric tests were made every half hour.

It was the intention in the long tests to begin and end with bare grates, but, owing to the position of the boiler in the vessel, and the small fire room, this was found impracticable, and the fire at beginning and end had to be "judged." This was done by three experienced officers, and is believed to involve only an inappreciable error.

The following abstract gives the results of the various trials:

Air pressure in inches of water	*0.0	0.5	3.0	4.0
Duration of test in hours	2.5	11.5	11.5	1.0
Temperature of feed	52°	58.5°	58°	52°
Steam pressure above atmosphere	250	250	250	250
Coal per hour per square foot of G. S	7.58	24.12	40 23	66.32
Refuse, per cent	6.11	8.04	7.53	6.27
Moisture, per cent	8.09	2.11	3.89	6.68
Apparent evaporation from temperature of feed at temperature of steam per pound of coal	10.55	8.13	7.53	5.69
Same from and at 212°	12.95	9.93	9.19	6.98
Actual evaporation from temperature of feed at temperature of steam per pound of coal	9.69	7.96	7.23	5.30
Same from and at 212°	11.90	9.72	8.84	6.51

During the first three hours of the test with three inches air pressure the combustion was at the rate of 59 pounds per square foot of grate.

The coal used in the experiments was Pocahontas of good average quality, as shown by the amount of refuse.

In view of the fact that Ward boilers are to be used on the *Monterey* and that Thornycroft boilers are in use on the *Cushing*, and that we now have accurate records of carefully conducted tests of both, made under the direction of the same distinguished engineer, it is very interesting to compare the performances of the two boilers under nearly the same circumstances.

It should be said that the men who fired on the *Cushing* were ordinary Navy firemen, with little or no previous experience in the use of forced draft, but the men who fired the Ward boiler were equally without experience of this kind. In both cases preliminary runs of several hours each were made to accustom the men to their work, and the observers, who recorded data, to their duties. The coal used with the Ward boiler was slightly the better.

COMPARISON OF THE PERFORMANCES OF WARD AND THORNYCROFT BOILERS TESTED UNDER THE DIRECTION OF CHIEF ENGINEER CHAS. H. LORING, U. S. NAVY.

	Ward.	Thornycroft.
Grate surface, square feet	53	38
Heating surface, square feet	2,473.5	2,451

* Blowers discharging into open fire room.

	Ward.	Thornycroft.
Ratio of H. S. to G. S.	46.67	64.5
Weight of empty boiler, in tons	11.84	9
Weight of boiler in steaming condition, in tons	13.85	11
Duration of trial, in hours	12*	11.5
Air pressure in inches of water	2	3
Feed temperature	50.4	58
Steam pressure above atmosphere	160	250
Coal per hour per square foot of grate	55.05	40.23
Refuse, per cent.	4.84	7.53
Moisture, per cent.	12.48	3.89
Apparent evaporation from temperature of feed at temperature of steam per pound of coal	6.60	7.53
Same from and at 212°	8.04	9.19
Actual evaporation from temperature of feed at temperature of steam per pound of coal	6.00	7.23
Same from and at 212°	7.31	8.84
Actual evaporation from and at 212° per hour per square foot of H. S.	8.62	5.51
I. H. P. per 100 square feet of H. S., on basis of 20 pounds of water per hour from and at 212°	43.10	27.55
I. H. P. per ton of boiler and water, on same basis, from performance of this boiler	76.98	61.38

It should be added that the test of the Ward boiler began and ended with bare grates, so that the comparison is not entirely fair. As shown in the last number of the JOURNAL (p. 416), the actual evaporation for ten hours, excluding the first and last hours, when the combustion was not as active as during the rest of the trial, was 7.72 lbs. per square foot of H. S., from 120° at 160 lbs., or 8.85 lbs. from and at 212°. This is an increase of about 2½ per cent., which would make the I. H. P. per 100 square feet of H. S., 44.25, and the I. H. P. per ton of boiler, 79.02, which are the figures that should really be used for comparison with the *Cushing* test.

ENGLAND.

Spanker.—"The new gunboat *Spanker* went out from Plymouth, on October 17, for a full-power forced-draft trial of her machinery. For over two hours the trial proceeded very satisfactorily, a speed of 19 knots being accomplished and the

* Mean of two trials of 12 hours each.

engines developing 3,800 I. H. P. Then the contractors tried to force the horse-power up to 4,000, but the boiler tubes began to leak. They leaked at such a rate that in a very short time the water level fell seriously, and there were quite three inches of boiling water running over the stokehold. The engine-room staff, with the greatest promptitude, pumped salt water into the boiler from the sea. The trial was at once stopped and the *Spanker* returned to port."—*Engineering*, October 24, 1890.

Barrosa.—This vessel, a sister ship to the *Barracouta*, on which such a serious accident to the boilers occurred on February 7, 1890, had several trials in the early part of October, which were interrupted by fog. One of these lasted, however, for 36 hours with an air pressure of one inch in the fire rooms. The trials would not be noteworthy except for the fact that several rows of tubes have been removed in the boilers to increase the circulation with a view to prevent the leakage at the tube ends, which has been such a serious defect in so many recent trials. It is very satisfactory to note that in this case no trouble whatever was experienced. It should be stated that these interrupted trials were intended to last 96 hours.

Although it is not stated what rows were removed, it is very likely that they were between and at the sides of the furnaces, so as to give room for a free down current of the water. For some time past it has been the practice of the Bureau of Steam Engineering to make such provision and to insure the thorough separation of the up and down currents by circulating plates secured to the sides of the nests of tubes. This has the effect not only of increasing and improving the circulation, but, as a consequence, increasing the evaporation by about twelve per cent.

Philomel.—A second-class, twin-screw cruiser, similar to the *Pandora*, was launched at Devonport, August 28. The hull dimensions are 265 feet × 41 feet × 15 feet 6 inches; displacement, 2,575 tons. The weight of the hull will be 1,030 tons; machinery, 540 tons; coal, 300 tons, and stores and equipment, 705 tons.

The machinery, by Earle's Ship and Engine Building Com-

pany, is of the horizontal, triple-expansion type. The cylinders are 30½, 45 and 68 inches diameter by 33 inches stroke. The I. H. P. under natural draft is expected to be 4,500, and under forced draft 7,500.

There will be four double-end boilers, 12 feet 6 inches diameter and 17 feet 3 inches long; working pressure, 155 pounds. The screws will be three-bladed, 12 feet 6 inches diameter and 14 feet pitch. The revolutions at 7,500 I. H. P. will be 160, giving 19 knots, and at 4,500 I. H. P., 140, giving 16 knots.

The armament will consist of eight 4.7-inch R. F. G., eight 3-pounder R. F. G., one 6-pounder R. F. G., and four four-barreled Nordenfeldts. There will be two torpedo tubes.

Area of plain sail, 5,030 square feet.

The total cost of the ship ready for sea is estimated at $710,000.

AUSTRIA.

Kaiserin Elizabeth.—This powerful protected cruiser was launched at Pola on the 24th of September. She is practically a sister ship to the *Kaiser Franz Josef I.*, which was launched at San Rocco, May 18, 1889. The hull dimensions are 339 feet over all by 47 feet 6 inches beam by 18 feet 8 inches mean draught. There is a protective deck fore and aft 2¼ inches thick, and a water-line belt of cellulose fiber. The twin-screw engines are expected to develop 6,400 I. H. P. under natural draft and 9,800 under forced draft, giving speeds respectively of 17½ and 19 knots. They are of the horizontal, triple-expansion type. There will be four double-end cylindrical boilers. The bunker capacity is 670 tons. There will be no sail power, but two military masts will be fitted.

The battery will consist of two 9.4-inch 27-ton Krupp rifles of 35 calibers on the upper deck, in barbettes 6 feet 4 inches high with 3½-inch steel armor. Between the barbettes is mounted a 5.9-inch 6-ton Krupp of 35 calibers. On the lower deck, in sponsons, are two similar guns on each side. There will also be two 2.7-inch 23-caliber Uchatius guns and eleven Hotchkiss R. F. G., besides six torpedo tubes, one in the bow right over the ram.

CHILI.

Almirante Condel and *Almirante Lynch.*—(Briefly noticed on page 237.) These torpedo gunboats have been built by Messrs. Laird Bros. of Birkenhead. They may be described as flush-decked vessels with high forecastle and half poop. The hull dimensions are 240 feet long, 27 feet 6 inches beam; draught forward 8 feet, and aft 11 feet, with a displacement at this draught of 750 tons. There is a steel ram stem and a bow torpedo tube.

On the forecastle are two 14-pound Hotchkiss guns *en échélon*, each gun having an arc of training of 220°. The conning tower on the after part of the forecastle is built of 1-inch plates and covered with a steel hood on which is a flying bridge carrying the search light and the mounting for two Gatlings. At the break of the poop, and on the main deck, is the armored hood containing an installation of engine-room telegraphs, steam steering gear, torpedo-firing and directing apparatus, etc., like that in the conning tower. This hood is capable of being inclined laterally so as to shield the officer beneath it. On the main deck are fitted four Canet torpedo-discharge tubes, two on each side. There are also two 3-pounder Hotchkiss guns. Water-tight subdivision has been carried out to the fullest extent, there being forty-five compartments.

There are four locomotive boilers, each with two furnaces of the ribbed pattern and water spaces under the ash-pits. The working pressure is 155 pounds. Each boiler is fitted with a non-return valve in its steam pipe, so that in the event of being pierced by a projectile it will be shut off from the others. The total grate surface is 190 square feet, and the total heating surface 6,000 square feet. Forced draft up to 2½ inches of water is provided by four fans.

The twin-screw engines of these vessels are duplicates of those for H. M. S. *Speedwell* and *Skipjack*. The cylinders are 22, 33 and 49 inches diameter, by 21 inches stroke. The valves are driven from Stephenson double-bar links. The H. P. valve is of the piston type, but those for the other cylinders are common slide valves. This necessitates large port area and a large valve.

The valve stems of the L. P. cylinders are prolonged, and are fitted with pistons which work inside small steam cylinders. Steam is admitted and exhausted by a small kicker slide valve. The result is that the work of moving the L. P. valves is taken off their eccentrics, which merely govern the various functions of opening, cut-off, &c. The reversing gear is combined hand and steam, of the "all-round" type.

There is, properly speaking, no bed plate for the engines, the framing and main bearings being carried on deep longitudinal girders secured to the framing of the ship.

The surface condensers, together with their tubes, tube plates and supporting diaphragms are of copper, one abaft each set of engines. The air pumps are vertical and single-acting, driven from the L. P. crossheads. The circulating pumps are centrifugal. The feed pump is worked, through worm gearing and an auxiliary shaft, from the same engine as the circulating pump. The revolutions are reduced in the ratio of six to one. There is in each engine room one of Kirkaldy's feed heaters and distillers.

The propellers are of manganese bronze and three-bladed. The diameter is 8 feet, the mean pitch 9 feet, and the combined developed area 42 square feet. They are cast solid, with a spherical boss. The crank and propeller shafts are of hollow steel.

Several preliminary trials have been made with both vessels, the results being uniformly satisfactory. The *Almirante Lynch* had an official trial September 25th, when a mean speed of 21.2 knots was easily maintained.

The *Almirante Condel* was tried on the 4th of October, with the following result as a mean of four runs over a measured course. The mean draught was 9 feet ½ inch, and the displacement 710 tons. Mean speed, 20.31 knots. Total I. H. P., about 4,350, with 270 revolutions as the average. Steam at boilers, 143 lbs.; at engines, 138 lbs.; first receiver, 46 lbs. (per gauge); second receiver, 8 lbs. (per gauge); vacuum, 27 inches; air pressure, 2¼ to 2½ inches.

The trial lasted for three hours, and as the weather was very

bad for so small a vessel, better results can be anticipated in smooth water. A coal consumption trial lasting six hours was made at speed of 11 knots, on October 7th, under a guarantee that at this speed the coal burned should not exceed 100 pounds per knot. It is stated that the actual result of the trial was an expenditure of only 88 pounds.

MERCHANT STEAMERS.

Plymouth.—This vessel is rapidly approaching completion for the famous Fall River Line of palace steamers, and while similar in many respects to the *Puritan* and *Pilgrim* is a radical departure as regards the machinery.

The hull is of steel throughout with a double bottom and six transverse watertight bulkheads. The dimensions are: length over all, 366 feet; length on water line, 351 feet 8 inches; beam, 50 feet; width over guards, 86 feet; mean draught, loaded, 12 feet; distance from keel to top of house on dome deck, 59 feet 4 inches.

The engine is designed for 5,000 I. H. P., and is of the triple-expansion type, the cylinders being arranged so as to form a double-inclined engine. The cylinders are one H. P., 47 inches, one I. P., 75 inches, and two L. P., each 81.5 inches diameter; the stroke of all being 8 feet 3 inches. The shaft is in three sections. Each outer section carries one crank arm to which the outer end of each of the two crank pins is keyed. These arms are the driving cranks. The central section carries the two other crank arms to which the inner ends of the crank pins are secured, the latter section with the arms acting as the drag cranks. The cranks are at right angles. The starboard crank pin has upon it two journals, and receives the connecting rods from the H. P. and one L. P. cylinder. The port crank pin has three journals. The connecting rod from the I. P. cylinder is

journalled to the central portion; the outer journals receive the connecting rod from the second L. P. cylinder, which rod is forked at the end. Owing to the inclination of the cylinders and the oscillation of the connecting rods, only one of the cylinders can be thrown on a dead point at a time. The power is therefore very well distributed and the shaft rotates with great regularity. The weight of the cranks, etc., would tend to disturb this regularity, but this is obviated by another peculiarity. The air pumps, of which there is one for each L. P. cylinder, are worked from the L. P. crosshead. As the cranks, connecting rods, etc., reach a position where their weight tends to accelerate the motion, the air pumps are working at their hardest, so that the overbalance is corrected.

The valves are vertical, balanced, double-poppet valves. For the H. P. cylinder an adjustable drop cut-off is used. The other cylinders have fixed Stevens cut-offs. The valves are worked by double eccentrics and links, the latter being shifted by a steam-reversing gear.

There is a condenser with air and centrifugal circulating pump for each L. P. cylinder. The combined cooling surface is 9,050 square feet. The air pumps are vertical, 40 inches diameter and 30 inches stroke.

The wheels are of the feathering type, 30 feet in diameter outside of the buckets. Each wheel has twelve curved steel buckets, each $\frac{1}{8}$ inch thick, 4 feet wide and 13 feet 3 inches long.

There are eight main boilers of 1 inch steel and cylindrical, 11 feet 4 inches diameter and 13 feet 1 inch long. The working pressure is 160 pounds. Forced draft can be used if desired. The smoke pipe is 10 feet 6 inches diameter, and extends 98 feet above the keel. The boilers are placed back to back in the center, leaving a fire room at each side. The coal is in bins above the fire rooms, to which it descends by gravity. The donkey boiler and powerful wrecking pumps are on the main deck, so that they could still be used with the main fire rooms flooded.

The contractors for the boat complete and builders of the engine are the W. and A. Fletcher Co., of New York. The hull

was built at the Delaware River Iron Ship Building Works, Chester, Pa.

Spree.—This vessel is the latest and largest of the North German Lloyd, and was built by the Vulcan Works of Stettin, Germany. Notwithstanding the great success of the recent fast twin-screw steamers, this company, like the Cunard, still cling to single screws, so that the *Spree* may be looked on as the latest and most improved of the single-screw vessels.

The hull is 485 feet long 52 feet beam, and 38 feet deep, with a registered tonnage of 9,000. The displacement is probably about 13,000 tons, on a mean draught of 26 feet 6 inches. The engines are vertical and triple-expansion with cylinders 38, 75 and 100 inches diameter by 6 feet stroke. There are ten boilers, six double-end and four single-end; diameter, 15 feet 6 inches; lengths, 18 feet 8 inches and 10 feet 4 inches, respectively. The working pressure is about 160 pounds. The propeller is four-bladed, of manganese bronze, 21 feet 7 inches diameter, and 31 feet 4 inches pitch.

On a trial trip in the Baltic, the *Spree*, though not worked up to her full capabilities, developed about 13,000 I. H. P. with 70 revolutions, giving about 20.1 knots.

OBITUARY.

JOHN HOWARD BAKER.

The career of this promising young officer was brought to a sad end by his sudden death from cholera at Chefoo, China, September 12, 1890.

His record was a creditable one in every way, professionally and socially. Great application and steadfastness of purpose made the satisfactory performance of any duty assigned to him an assured fact. His personal qualities, in which cheerfulness

amounting almost to optimism was a strong ingredient, made him an always pleasant shipmate and he was a universal favorite.

He was born at Cleveland, Ohio, August 31, 1857; appointed a Cadet Engineer at the Naval Academy, September 15, 1875. His service at sea was on the *Standish*, *Nipsic*, *Trenton*, *Enterprise*, *Essex*, *Swatara* and *Monocacy*, to which last he was attached at the time of his death.

He was commissioned an Assistant Engineer, June 10, 1881.

His only shore duty was at St. John's College, Annapolis, as Professor of Mechanical Engineering, from 1885 to 1888.